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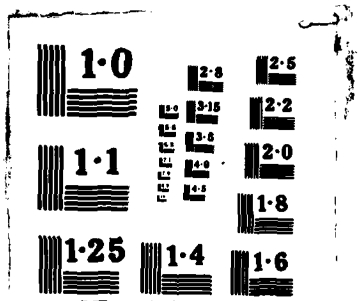
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AIRCRAFT GEAR AND BEARING TRIBOLOGICAL SYSTEMS

Papers presented at the 60th Meeting of the Structures and Materials Panel of AGARD
in San Antonio, Texas, USA, on 22–26 April 1985.

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PREFACE

Tribology, in its simplest sense, is the science that deals with the friction, wear, and lubrication of interacting surfaces in relative motion. In a broader sense, it includes all aspects of the design, engineering, operation and maintenance of mechanical systems involving interacting surfaces, including such considerations as materials development and application, failure mechanisms, system diagnostics or health monitoring, and repair or maintenance procedures. As an activity of the Structures and Materials Panel of AGARD-NATO, this Subcommittee and the associated Specialists' Meeting are concerned primarily with materials and structures used in gear and bearing systems of aircraft, including both fixed-wing and rotary-wing types. Other Specialists' Meetings addressing a broader range of operational and total system performance considerations have been sponsored by the Propulsion and Energetics Panel.

Aircraft gear and bearing tribological systems are receiving increased attention throughout the entire NATO community. Two main factors are driving this emphasis:

1. the need for increased performance and reliability in both military and commercial aircraft, and
2. the need to reduce both acquisition and total life cycle costs of military and commercial aircraft.

The cost and performance of gear and bearing systems are important factors in the overall cost and performance of all modern aircraft. This is particularly true for helicopters, where the power transmission systems are major contributors to the total weight and cost of the aircraft, but it also applies to high-performance fixed-wing aircraft, where relatively small changes in component performance, weight and cost are more critical.

The drive for higher performance in aircraft gear and bearing systems centers on the three principal goals:

1. reduced weight
2. higher operating speeds
3. higher operating loads

All three of these goals could, in principle, be accomplished through the use of higher strength steels that would permit higher operating stresses in shafts and gear teeth and higher compressive and Hertzian shear stresses at the interacting surfaces of gears and bearings. The situation is not quite that simple, however. Higher loads and speeds, and reduced contact areas lead to higher heat generation, which can lead to scuffing of the faying surfaces as well as accelerated deterioration of the lubricant or even complete breakdown of the lubrication system. Also, most higher strength steels possess lower fracture toughness than conventional gear steels, so that care must be taken to avoid catastrophic gear and bearing failure. It is thus a complex problem, but improved gear and bearing materials, improved lubricants, and novel design approaches could bring about substantial improvements in gear and bearing systems performance and consequent overall aircraft performance.

Cost reduction has become an overriding consideration in the design, development, and procurement of both military and commercial aircraft in most NATO countries as a result of popular demands for reduced military spending and increased competition in both aircraft and airline industries. Gear and bearing systems not only constitute a substantial portion of the acquisition cost of aircraft, but are also a major source of repair and maintenance costs during the life of the aircraft. These costs, which are frequently many times higher than the original acquisition cost of the systems, include not only the cost of spare parts and labour to install them, but also the cost of the entire logistic support system required to maintain the fleet. An even larger cost is that associated with shortened life of the aircraft and their non-availability during periods of repair and maintenance. Most repair and maintenance costs in aircraft gear and bearing systems result from deterioration or failure of the gear and bearing elements themselves by one of three processes:

1. wear
2. corrosion
3. fracture

These deteriorative processes are strongly influenced not only by the overall system design and operating conditions, but by the nature and properties of the gear, bearing and lubricant materials themselves.

In this Specialists' Meeting we have learned of many recent advances in the broad field of aircraft gear and bearing tribology:

- New analytical methods and design concepts are being developed.
- New gear and bearing steels with improved hot hardness and toughness have been developed and evaluated.
- New surface treatments such as vacuum carburization and ion implantation have been demonstrated.

- Ceramic bearings have been developed and are under evaluation.
- New synthetic lubricants with improved performance and resistance to deterioration are under development.
- New corrosion prevention and control procedures are being developed and employed.
- New system health monitoring and maintenance procedures are now available.
- A better understanding of the science of tribology and the complex interaction of machine elements, lubricants, and the operating environment is rapidly evolving.

In summary, it appears that we are on the verge of a great advance in aircraft gear and bearing system technology. The challenge is to combine all of the emerging developments in the various disciplines that compose the complex field of tribology, and focus a coordinated international effort to bring about a major advance in the state of the art.

Edward S.Wright, Chairman
SMP Subcommittee on
Aircraft Gear and Bearing Tribological Systems

CONTENTS

TRIBOLOGICAL SYSTEMS AS APPLIED TO AIRCRAFT ENGINES

by
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SUMMARY

Tribological systems as applied to aircraft are reviewed. The importance of understanding the fundamental concepts involved in such systems is discussed. Basic properties of materials which can be related to adhesion, friction and wear are presented and correlated with tribology. Surface processes including deposition and treatment are addressed in relation to their present and future application to aircraft components such as bearings, gears and seals. Lubrication of components with both liquids and solids is discussed. Advances in both new liquid molecular structures and additives for those structures are reviewed and related to the needs of advanced engines. Solids and polymer composites are suggested for increasing use and ceramic coatings containing fluoride compounds are offered for the extreme temperatures encountered in such components as advanced bearings and seals.

INTRODUCTION

Considerable research effort has been put into advanced aircraft systems over the past 50 years, however, major significant accomplishments have been relatively few over the past 20 years. Most of the advances have been incremental improvements in systems, components and materials. There is a need to develop a visionary spirit which transcends the examination of the next generation aircraft while still tending to current demands. This need exists for the entire aircraft but particularly the tribological systems and even more specifically those associated with engines.

Many exciting advances have been made in the past 20 years in our understanding of materials, for example, and their interactive behavior. All too frequently the materials, particularly those associated with tribological applications are not given adequate consideration in initial design stages. Rather than incorporating new materials and application concepts into design the more conservative approach of making materials selections from the present inventory has been the choice.

A sound fundamental understanding of tribological concepts and some of the opportunities available to the designer could offer the prospect for future significant advances. Once a basic understanding is achieved as to why certain materials behave as they do and what properties of materials are important to tribological behavior the tools are in hand for these advances.

The objective of this paper is to: (1) review fundamental properties of materials important to adhesion, friction and wear, (2) examine properties important in the material selection process for tribological applications, (3) address lubrication regimes and new lubricants, and (4) examine surface treatments that may be incorporated in current and more particularly advanced aircraft engine bearing and gear components. The goal is understanding rather than specific recommendations.

FUNDAMENTAL TRIBOLOGICAL CONCEPTS

Understanding the adhesion, friction, wear and lubricated behavior of solids in contact is extremely important to understanding bearing, gear and seal performance in aircraft engines. It provides the basis for material selection, design considerations, lubricant use and material treatments.

Figure 1 schematically depicts two solids in contact as might occur in bearings, gears or seals. Real area of asperity contact is depicted at the interface by the black patches. The lesson to be remembered from this figure is that the load in mechanical components is borne by these areas and not the apparent contact area.

A second observation to be made for Fig. 1 is that when any two surfaces can come into intimate solid state contact strong bonds of adhesion can develop and these bonds effect friction. The stronger the bond, the higher the friction and the less efficient is the tribological component.

Yet a third point to be made with Fig. 1 is that associated with the wear of the two solid surfaces. There are various mechanisms of wear, adhesion, abrasion, corrosion, cavitation, erosion and fatigue all of which occur and are important to aircraft tribological components. These mechanisms will be discussed in the next section of this paper.

The final area to be considered in Fig. 1 is that associated with reducing, adhesion, friction and wear and thereby improving life namely lubrication. The objective of

lubricating is to keep the surfaces from interacting and forming surface bonds. The lubricants as will be discussed in a latter section can be liquids or solids. The more effective they are in keeping the surfaces separated the better they are as lubricants.

MATERIAL PROPERTIES

There are a host of fundamental properties of materials which are related to tribological performance (Ref. 1). The designer by keeping these in mind while selecting materials can do much to optimize the materials selection process.

Some recent advances in material properties, by way of example, which can be related to tribological performance are indicated in Figs. 2 to 5. They demonstrate fundamental relationships to particular tribological properties.

Figures 2 and 3 deal with the ever present fretting problem encountered in aircraft engine components. Fretting is a result of two wear mechanisms operating, adhesion and corrosion. Recent studies indicate that certain bulk and surface properties of metals can be related to fretting (Ref. 2).

In Fig. 2 the fretting wear volume of pure metals is related to the shear strength of the metal. The higher the shear strength the less is the fretting wear. The selection of materials for aircraft components where fretting may be a problem should factor in this relationship.

The data of Fig. 2 indicates the adhesion component of fretting, namely the higher the shear strength the more difficult it is to generate an adhesive wear particle. The data of Fig. 3 reflects on the corrosive component of fretting.

The data of Fig. 3 indicate a correlation between fretting wear at high humidity (RH_{max}) and the heat of oxygen adsorption on the metal. The stronger the oxygen bond the lower is the material loss due to fretting. In a strict sense the adsorbed species are acting as a lubricant preventing adhesive wear from occurring. Remember what was said earlier relative to lubricant effectiveness. The better able the lubricant is to separate surfaces the more effective it is as a lubricant. Strong bonding, of course, reflects tenacity.

From a consideration of the data of Figs. 2 and 3 the designer with a fretting problem or potential one is better able to make a material selection. At least there are some guides provided.

Another form of wear frequently encountered in aircraft component design is that of abrasive wear. Abrasion can cause the rapid loss of material from bearing, gear and seal surfaces resulting in frequent component replacements. Are there basic guides to assist the designer with combating this form of wear? There certainly are properties directly relatable to abrasion. The most well know is that of hardness. The harder the material the greater is the abrasive wear resistance (Ref. 3). This has been known for some time as reflected in the year of Ref. 3. There are more recently studied properties of materials which can be related to abrasive wear and to friction. One of these concerns fundamental alloy concepts (Ref. 4).

An examination of simple binary alloys indicates that there is relationship between an atomic property, namely lattice radius ratios and abrasive wear. Further, it establishes an inverse relation to friction. The friction data of Fig. 4 is for silicon carbide abrading simple iron binary alloys.

The data of Fig. 4 indicate that during abrasion friction is at a minimum where the atomic ratio of the alloying element to iron is unity. Deviation to either side in size results in increased friction, increased hardness and increased wear resistance.

Another common form of wear encountered in aircraft, particularly engine components is that of erosive wear or erosion. While it does not generally cause incipient component failure as can be experienced with adhesion it limits component life. Are these any basic material properties that we can relate to this form of wear? Again recent fundamental studies indicate that there are such properties (Ref. 5).

The data of Fig. 5 indicate a correlation between melting point of metals and their erosion resistance for various erodents, particle sizes, particle velocities and angles of impingement. Other properties, such as surface energy, strain energy, bulk modulus, hardness, ultimate resilience and atomic volume have also been correlated with resistance to erosion (Ref. 5).

BEARING, GEAR, AND SEAL MATERIALS

Through the years a host of alloy materials have been employed for rolling element bearings and a number have been used in aircraft applications. Table I presents some representative steels. The currently most popular material is, of course, M-50.

An interesting observation to be made from the list in Table I is that not a single one of the alloys was specifically uniquely designed in its composition for aircraft engine bearing applications. Through a process of extensive and expensive empirical testing M-50 has by and large been the survivor. But what about the future? Are

engines going to run on M-50 henceforth? Insight is needed to formulate new bearing materials for future aircraft.

In deciding a new bearing composition for future aircraft applications thought must be given to what properties this new material should have incorporated into its structure. Table II presents some requirements for future bearing materials. With a knowledge of the limitations of M-50 and the areas in need of improvement from Table II there is the basis for beginning of selecting property requirements of a new alloy and formulating its composition. It will for its development require the joint effort of both the designer and materials engineer. In the interim some of the surface treatments discussed in another section of this paper may offer improvement in the performance of existing bearing materials.

Table III presents some of the more commonly used gear materials in aircraft systems. In addition gears have been made from a host of material classes including polymers, brasses and bronzes, sintered powder-metal alloys and other steels. Again, as with bearings there are certain requirements for advanced gears materials as indicated in Table IV. With gears, processing technology appears to be the current greatest impediment to significant advances.

Seal materials in aircraft are required for lip, ring, face, and labyrinth seals. Rubbers, acrylates, silicones, fluoroelastomers, and fluorocarbons (TFE) have all been used in lip seals. Higher temperatures in advanced engines will require materials with greater thermal and oxidative stability.

Today aircraft gas turbine mainshaft and accessory gear box ring and face seals are carbon-graphite structures. While these materials can be improved upon, new more oxidation resistant materials should be examined including ceramics such as silicon nitride and carbide, refractory metal alloys and coatings.

With labyrinth seals current technology involves the use of solid metal knife edges where the opposing lands are solid metal or abradable coatings applied to nickel alloy substrates. There is considerable room for improvement with the application of composite structures.

SURFACE MODIFICATION TECHNIQUES

In recent years considerable advances have been made in the deposition of protective surface coatings and treatments of surfaces to modify various properties. Coatings and surface treatments have been employed for altering catalytic behavior of surfaces, reducing corrosion, altering surface hardness, reducing friction, extending life, and the reduction of wear. Many of the techniques offer considerable promise for use in aircraft tribological components such as bearings, gears, and seals.

The principal recent advances in coating technology have resulted from the use of plasma physics deposition techniques. They have allowed for the deposition of soft metal films and soft inorganic compounds such as molybdenum disulfide for solid film lubrication, hard face coatings such as carbides and nitrides for higher hardness and improved wear life and noble metals for catalysis and corrosion protection.

Surface treatments involve the use of beam energy sources such as laser, ion, and electron. A variety of surface property changes can be induced with these treatments. Figure 6 presents schematically both the coating and treatment approaches. With coatings such processes as ion plating, sputter deposition, and chemical vapor deposition (CVD) offer protective surface films to reduce adhesion, friction, and wear as well as extend the life of mechanical components.

Ion plating provides an ideal process for achieving both corrosion and tribological protection. The process allows for the application of extremely thin (1500 A) films of high density, uniformity in thickness (50 A) with a diffuse or graded interface for maximum interfacial adhesion. It is especially adapted for the deposition of metallic films and alloys with not too greatly different vapor pressures for the alloying elements. The coating material is brought to the surface in a flux of argon ions and the coating materials consists of a mixture of atoms and ions. They penetrate the surface of the negatively charged substrate to be coated providing for the diffuse or graded interface between coating and substrate.

Sputter deposition allows for the application of nearly any type of coating, polymer, metal, alloy, ceramic or inorganic solid lubricants to nearly any substrate. The incoming flux to a surface can consist of ions, atoms, molecules, and molecular fragments. It is, for tribological applications, particularly useful in applying polymeric films of materials such as polytetrafluoroethylene and solid film lubricants such as molybdenum disulfide. While complex geometric surfaces such as small gear teeth can be coated with sputter deposition it does not have the "throwing power" of ion plating (ability to get into complex configurations). Further, the interface between coating and substrate is not diffuse or graded as with ion plating. Its principal merits lie in the ability to apply a wide variety of coatings to a host of substrate materials.

Chemical vapor deposition (CVD) has been introduced into tribology primarily with the deposition of hard face coatings for improved wear resistance and longer wear life. Refractory metal carbides, borides and nitrides have been deposited by this technique.

Gaseous carriers are employed to bring components of the coating material to the substrate. For example, if one were to desire a silicon nitride coating, two gaseous species would be admitted into the plasma silicon tetrachloride ($SiCl_4$) and either nitrogen (N_2) or ammonia (NH_3). Balancing gaseous ratios is important in achieving proper coating chemistry. One disadvantage of the process is that frequently the substrates must be heated to high temperature (i.e., 500 °C) during deposition. In coating heat treated bearings and gears such temperatures could destroy prior heat treatments.

With respect to surface treatments laser glazing is a relatively recent approach to altering near surface metallurgy and topography and it has considerable merit. Using a laser beam one can produce a smoothing in bearing or gear surface topography, heal surface defects such as microcracks, particularly those produced in production. It also can by rapid heating of the surface to melting and rapid quenching be used to generate "amorphous" surface layers. Wear studies on "amorphous" metals or metallic glasses indicate that the absence of crystal structure can provide superior wear resistance for such applications as foil bearings.

Ion beam treatments of surfaces can include ion etching, ion nitriding and ion implantation. In all cases a beam of ions is directed at the surface. The specie and the energy varies.

In ion etching a beam of inert gaseous ions will be directed at the surface to remove surface layers. It could be used to remove processing defects, undesirable surface contaminating layers, produce carbides in relief and roughen the surface, without introducing stresses, for the purpose of generating lubricant reservoirs.

With ion nitriding nitrogen ions are directed at the surface. They interact with nitride forming elements present at the tribological surface to produce the nitrides. It is a "clean" process in that it is conducted in a vacuum system with pure nitrogen gas.

The ion implantation process is capable of implanting ions of a desired species into surficial layers of the material. It is a high energy process with ion energies from 10 to 200 keV. Gaseous species such as nitrogen have been implanted into bearing materials as well as metallic ions such as titanium.

Electron beams and the energy associated therewith have been used to produce surface heating without raising the entire component to some desired surface temperature. It also is effective in altering or modifying polymeric material surfaces. The electrons can serve to produce bond scission with recombinations in structures that vary in composition from the original material.

LUBRICATION

All components of aircraft requiring lubrication are in fact lubricated by one of four principal lubrication regimes. With liquids these regimes are as indicated in Fig. 7. When the surfaces are separated by thick films with shear in the fluid as in journal bearings the hydrodynamic regime of Fig. 7 is operating (Ref. 7). In this regime the fluids viscometric properties and oxidative and thermal stability are important as are the wetting and acceptability to the surface of additives. In looking to future aircraft applications those fluids offering high temperature oxidative and thermal stability must be examined. To that end research is presently being conducted with the fluorinated polyethers and the temperature range of stability is indicated in Fig. 8. These fluids offer promise to 315 °C but to achieve even higher temperatures more research will have to be expended on materials such as the fluoroether triazines which as indicated in Fig. 8 have potential for usefulness to 350 °C.

With new fluids arises the need for additives which are compatible with these new molecular structures. One additive used in many fluids is the antioxidant. This additive is particularly important with fluorinated fluids such as those indicated in Fig. 7. Studies with phosphorus containing additives indicate oxidative degradation of these materials can be considerably arrested by the use of such additives (Ref. 8).

Figure 9 presents data for the oxidative degradation reactions of perfluoroalkylethers with and without oxidation inhibitors. Even in the presence of active titanium alloys degradation of the fluid lubricant can be appreciably retarded. All three additives of Fig. 9 were effective in reducing the breakdown of the perfluoroalkylethers in the presence of oxygen. For advanced aircraft systems considerable research is needed to explore additive surface interactive chemistry in order that optimum additive selection is made for advanced fluids.

In the elastohydrodynamic regime of Fig. 7 many of the same concepts already discussed in reference to hydrodynamic lubrication apply. The rheology of the fluids become extremely important in this area so important to rolling element bearings. While EHL theory is well in hand there is a strong need for the incorporation of real surface effects into the theory. Fluid temperatures in this regime have been experimentally measured and reach 350 °C and the topography of the bearing surface has been shown to be extremely significant. These real effects must be factored into future analysis.

Of all the regimes indicated in Fig. 7 the one that has received the least attention and that is in most need of understanding is the mixed film regime between boundary and elastohydrodynamic lubrication. What happens when the EHL films breakdown and solid

state contact occurs. This certainly happens in aircraft components but has not been addressed with sufficient effort to arrive at an understanding.

It can be stated with certainty that the regime of Fig. 7 having received the greater attention is that of boundary lubrication and rightly so. It is in this regime that solid state contact occurs and the friction and wear reducing properties of the lubricant become all important.

First, there is the lubricating and stability properties of the fluid where liquid lubrication is employed. In most aircraft applications it is not so much the base fluid that is depended upon but rather the additives in the fluid for boundary lubrication. The base fluids thermal and oxidative stability are extremely important as the fluid must carry the additives to the surfaces needing lubrication and carry away heat.

In the search for new fluid lubricants for advanced aircraft systems the stability of the molecule must be examined. A good guide in examining structures is to determine the weakest bonding in the molecule and establish the dissociation energy of that bonding. This will provide insight into the thermal stability of the fluid as indicated in Fig. 10.

The data of Fig. 10 indicates the stronger the bond energy the higher the decomposition temperature (Ref. 9). Using this approach one can verify that the fluoroether triazines of Fig. 8 have thermal stability to 350 °C.

As already mentioned it is in the boundary regime where additives are so all important in keeping the surfaces in solid state contact from wearing. With advanced aircraft design attempts to increase the load carrying ability of mechanical components will require both antiwear and extreme pressure (EP) additives superior to those presently in use. Figure 11 indicates how these additives effect both load carrying ability and wear in the boundary lubrication regime (Ref. 10).

From an examination of Fig. 11 it is readily apparent that improvements in wear (ΔK) can be achieved with antiwear additives while the load carrying ability (ΔF) of a base oil can be markedly improved with the proper extreme pressure additives). Considerable research is currently being conducted into the mechanism of EP additive lubrication so that new and superior additives will be available for advanced lubricants.

There are those components of aircraft which can not be lubricated with conventional fluid film lubrication, for example, where the temperatures exceed the 350 °C discussed earlier. Under such conditions solid films with lubricating properties are employed. Lubrication with these materials are in the boundary regime because solid state contact is continuous.

Some of the most successful solid film lubricants have been the dichalcogenides, MX_n compounds of S, Se or Te with a hexagonal-layered crystal structure. The easy shear parallel to the basal planes of the crystallites make them ideal solid lubricants. Some of these materials and their properties are presented in Table V.

The data of Table V indicates that these solids have application in aircraft parts at temperatures well above that experienced for liquids (Ref. 11). These materials are, however, just as are fluids, sensitive to environment. This sensitivity is demonstrated in the data of Fig. 12 for molybdenum and tungsten disulfides (Ref. 12).

In argon both compounds in Fig. 12 exhibit low coefficients of friction to 400 °C and acceptable values to 1000 °C. When the atmosphere is air, however, at 400 °C for the molybdenum disulfide and 600 °C for the tungsten disulfide friction increases appreciably due to the oxidation of these compounds to their respective oxides. When designing components for aircraft employing these materials the designer must maintain cognizance of these sensitivities particularly since these materials are seeing ever increasing use in aircraft.

A very promising lubrication system for advanced aircraft is to make greater use of polymer composite structures (Refs. 13 to 19). These materials provide self-lubricating structural members. Various fillers can be incorporated into a basic polymer structure such as a polyimide. The filler can be metal, glass or graphite either powders or fibers.

Figure 13 presents an example of the application of a polymer composite to a spherical bearing. The composite as indicated in the figure may be used as the ball (upper schematic) or it can simply be used as a liner (lower schematic). There are broad opportunities offering considerable promise for the use of such materials in future aircraft systems where liquids can not be effectively used.

For future applications such as labyrinth seals where abradable seal materials are used and temperatures of rubbing surfaces are extremely high it may be necessary to lubricate with ceramic coating that have built in lubricating properties. Some of these materials have been successfully used in bearings at temperatures to 900 °C (Ref. 20).

Figure 14 presents friction data for two of these compositions containing, nichrome, calcium fluoride, glass and one containing the addition of silver to reduce friction coefficient at lower temperatures. They can provide effective lubrication to 900 °C as indicated in the data of the figure.

SUMMARY REMARKS

A review of the present state of the art in tribological systems for aircraft engine applications indicate that there have been many recent advances in our understanding of materials and lubrication which can assist in improving tribological performance. If the design engineer gains a better fundamental understanding of the properties of materials which are related to adhesion, friction and wear more judicious selection of the appropriate materials can be made for bearing, gear and seal applications.

The significant progress in surface treatment technology offers a variety of opportunities in extending tribological component life. The use of plasma assisted deposition processes allow for thin films to be deposited on complex geometries, with uniformity, high density and with outstanding tenacity. Soft metals, polymers and inorganic compounds can be deposited for lubrication, hard face coatings for wear resistance and surface treatments such as ion nitriding for extended life.

In the area of lubrication new fluids and additives are being studied to extend the useful temperature range of liquid lubricants. Synthetic molecular structures are being examined for both oxidative and thermal stability and additives compatible with these fluids. Where liquids can not be used because of temperature limitations solids, such as soft metals and diechalcogenides are employed as well as polymer and polymer composite structures. For the extreme temperatures, beyond 500 °C, ceramic coatings containing fluorides have been shown to be effective.

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TABLE I. - BEARING MATERIAL

AISI M-50	AISI 9310 (C)[a]
AISI M-10	CBS 600 (C)[a]
AISI M-1	CBS 1000M(C)[a]
WB-49	CRB - 7
AISI 440-C	
SAE 52100	
AMS 5749	

[a](C)-carburized grades.

TABLE II. - REQUIREMENTS FOR FUTURE BEARING MATERIALS

Improved wear resistance
Improved corrosion resistance
Improved fracture toughness
Present hot hardness
Improved fatigue strength

TABLE III. - GEAR MATERIALS

AMS (AISI 9310)*
CBS 600
VASCO X2
PYROWEAR 53

TABLE IV. - REQUIREMENTS FOR FUTURE GEAR MATERIALS

High hot hardness
Improved fatigue life
Better fracture toughness
Superior wear resistance

TABLE V. - RESULTS OF THERMAL STABILITY AND FRICTIONAL EXPERIMENTS IN VACUUM OF 10^{-9} to 10^{-6} torr

Compound	Probable onset of thermal dissociation as detected by TGA, °C	Dissociation products first detected by mass spectrometry, °C	Maximum temperature at which burnished film provided effective lubrication, °C
MoS_2	930	1090	650
WS_2	870	1040	730
$MoSe_2$	760	980	760
WSe_2	700	930	760
$MoTe_2$	700	700	540
WTe_2	700	700	(a)

[a]Friction coefficient greater than 0.2 at all temperatures.

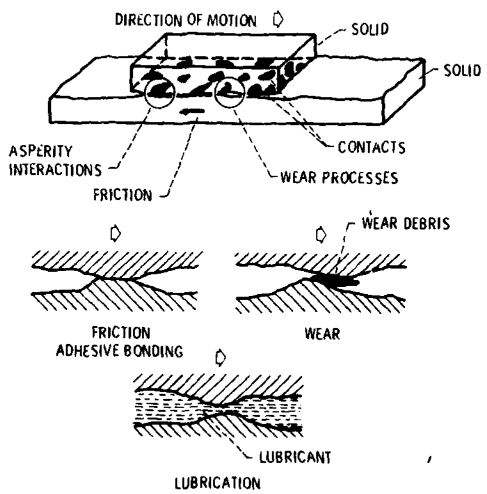


Figure 1. - Tribological properties of materials (adhesion, friction, wear and lubrication).

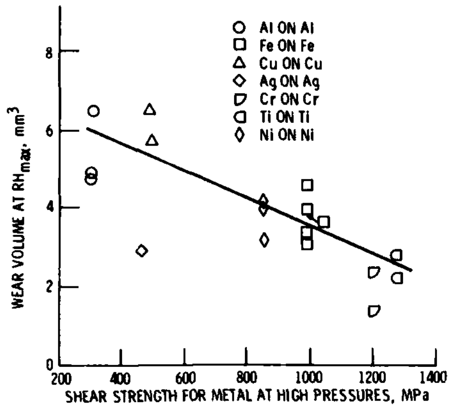


Figure 2. - Fretting wear volume at RH_{max} as a function of shear strength for pure metal at high pressures.

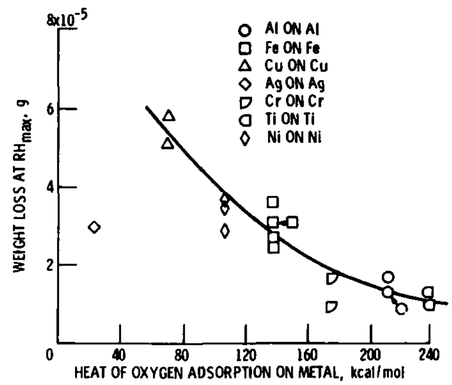
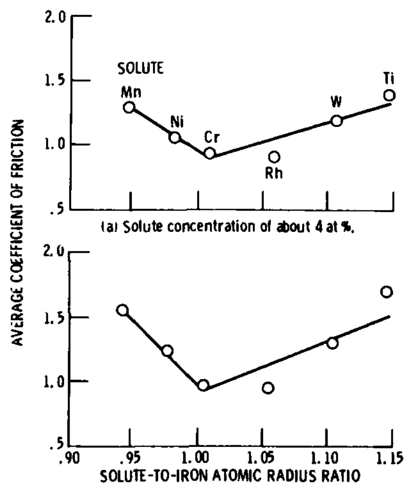


Figure 3. - Weight loss due to fretting wear at RH_{max} as a function of heat of oxygen adsorption on metal surface.



(a) Solute concentration of about 4 at %.

(b) Maximum solute concentrations of alloys.

Figure 4. - Coefficients of friction for iron-based binary alloys as function of solute-to-iron atomic radius ratio. Single-pass sliding on single-crystal silicon carbide (0001) surface.

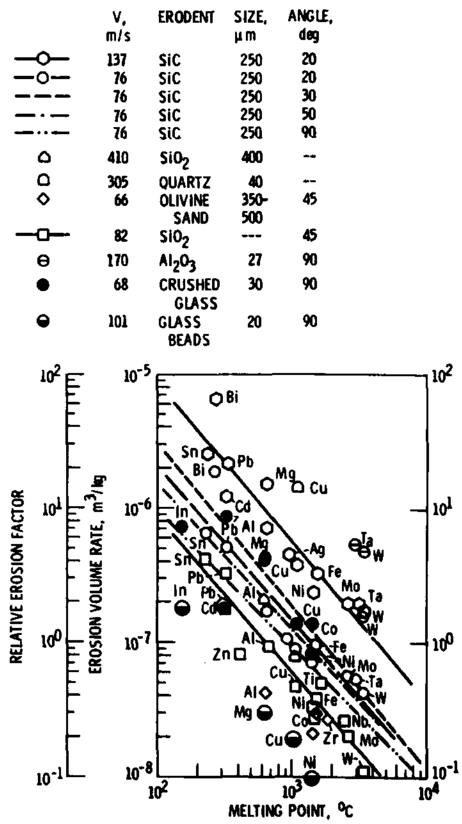


Figure 5. - Erosion rates of different metals as a function of melting point.

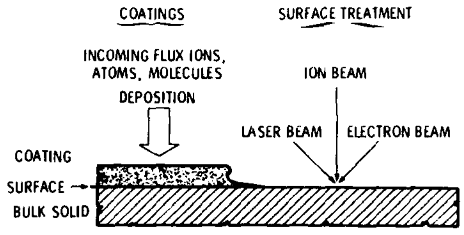


Figure 6. - Surface modification techniques.

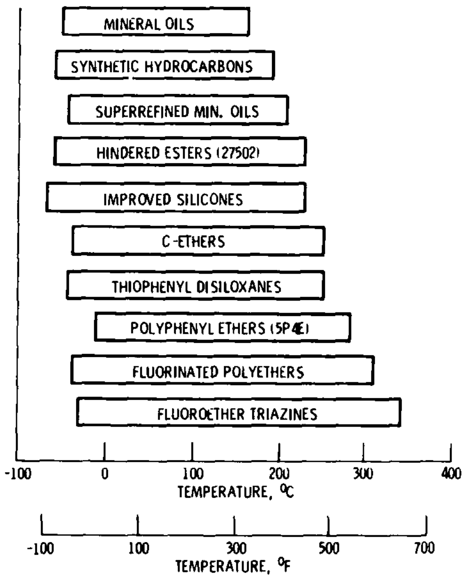


Figure 8. - Operating temperature range for classes of high temperature lubricants.

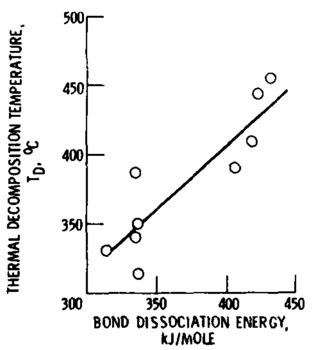


Figure 10. - Thermal decomposition temperature (T_d) as a function of bond dissociation energy.

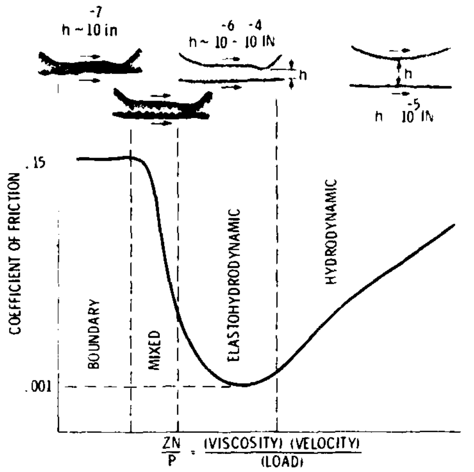


Figure 7. - Coefficient of friction as a function of speed-velocity-load parameter (Stribeck-Hersey curve) (ref. 1).

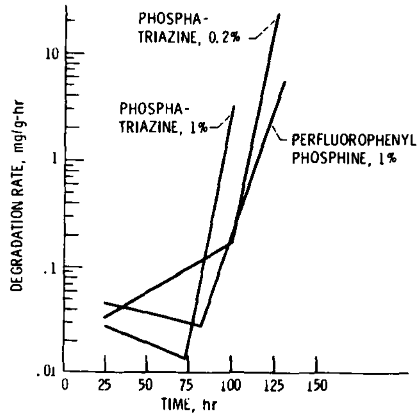


Figure 9. - The effects of metals and inhibitors on thermal oxidative degradation reactions of unbranched perfluoroalkylethers (288 °C, O_2, Ti (4Al, 4Mn) alloy).

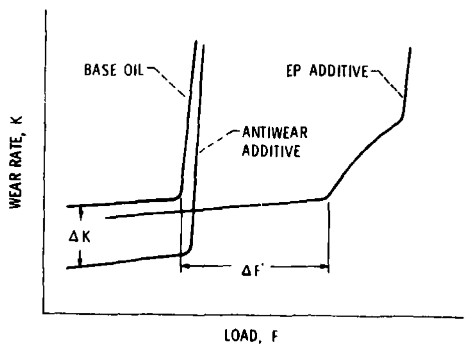


Figure 11. - Wear behavior of boundary-lubrication systems (ref. 10).

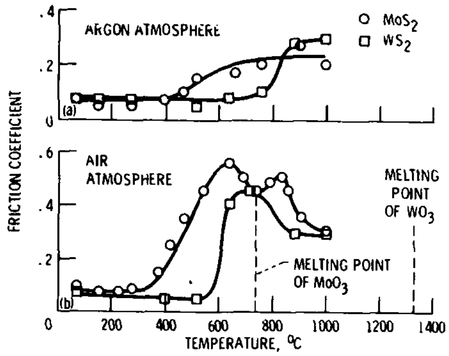


Figure 12. - Friction characteristics of MoS_2 and WS_2 in argon and in air.

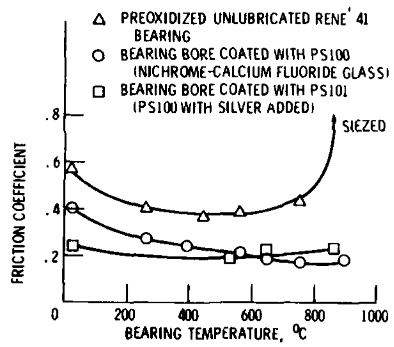


Figure 14. - Bearing friction.

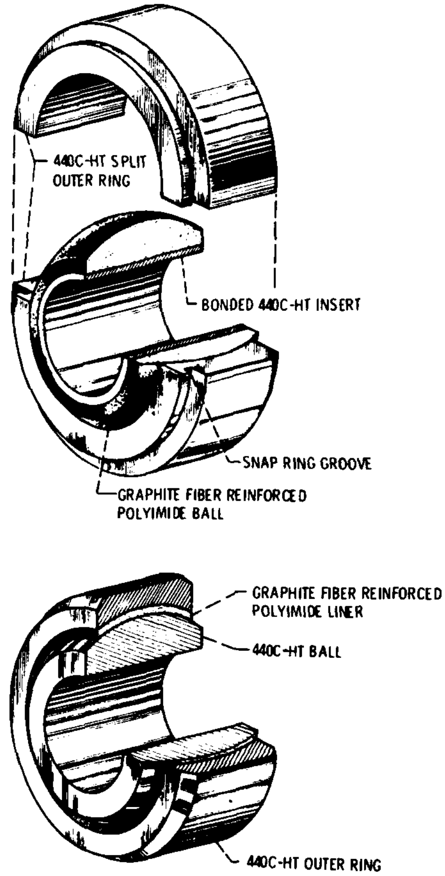


Figure 13. - Test bearings employing graphite fiber reinforced polyimide.

AD-P005 053

The Value of Laboratory Simulation Testing for Predicting Gearbox Performance

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SUMMARY

The evolution of a successful gearbox design involves much development testing which is both time consuming and expensive. For this reason full advantage should be taken of laboratory simulation tests for which, to be successful, predictions must, obviously, be reliable. Some of the more salient aspects of simulation testing are set out along with a few examples drawn from the author's experience.

INTRODUCTION

The time remains distant before designing a gearbox analytically can become possible. At present a designer draws heavily on experience and then development takes over. This slow and costly procedure has usually only been partially completed when the design has to be frozen for production. From that stage onwards only essential modifications are introduced and, hence, it is the norm for gearboxes to enter service only partly developed. No opportunity should be missed that can lead to a reduction in development time.

In practice no design rules exist, other than empirically derived guidelines, to enable a designer to predict scuffing, pitting and fretting performance when using modern lubricants. He has to lean heavily on laboratory simulation testing which, if reliable can play a major role in reducing development times. Inevitably it is left to the laboratory to guess those essential parameters that are to be simulated. Are gear localised contact stresses and flank temperatures ever known in an existing design let alone realistically established for a new one? The answer is an emphatic no. Methods have now been developed for monitoring surface temperatures, contact pressures and oil film thicknesses in both bearing and gear tooth contact so, looking towards the future, much of the unnecessary guesswork can be eliminated. However, much guesswork still remains and for many years to come this is likely to remain the position. There are too many variables in the make up of both the oil and the steel for 'Building Block' models to provide other than general guidance.

There can be no substitute for actual gearbox testing and this should run parallel with recognised laboratory testing, such as disc machine evaluations, in order to provide the necessary background experience to make practical predictions from laboratory tests. This is not, however, the complete answer. Gearbox testing usually is performed in a "Back to Back" rig or, occasionally against a brake. Rig performance does not coincide with that experienced when the gearbox is operated in a vehicle no matter how carefully the operating role be simulated and allowance made for differences in vibration characteristics. The starting point is the thorough surveys of gearboxes returned for overhaul, repair, or defect reporting. Armed with this background information relationships can be established between gearbox testing in flight simulation roles and actual helicopter performance. Correlationships between the range of laboratory and gearbox testing become realistic with the necessary background experience until the stage is reached when even the most elementary form of laboratory test can usefully be used for making certain full scale predictions.

It is not, however, a simple matter of simulating full scale conditions and reading off results. It is the sensible interpretation of such results that is of paramount importance. Clearly laboratory test machines should relate as closely as possible to the conditions to be simulated. The design of suitable machines is a neglected area, so much so that virtually all the ones with which the author has been involved had to be designed specifically for the task in hand.

Rather than simply set out a series of examples showing how closely predictions can relate to full scale performance it is thought that a few observations would be of more interest. Predictions still come within the category of an art, supported by science, so it is not possible to lay down rules, only guidelines.

SIMULATION REQUIREMENTS

It will be convenient for present purposes to deal only with scuffing and pitting because these form the basis of most oil/metal evaluations. Precise full scale conditions, usually, cannot be reproduced and thus the key to successful prediction lies in knowing what the differences are and how to account for them afterwards.

Test times clearly have to be kept down to a practical minimum and this usually leads to some form of accelerated testing by running at higher loads or speeds or at a different temperature. The important issue here is that subsequent extrapolation to working conditions must be realistic. Continuous monitoring of all relevant parameters is more important than may be apparent, especially when the unexpected happens. The following list of main points have to be considered.

1. Speed: Controls oil film thickness, both directly and through surface temperature and, hence, λ values.

2. Surface Temperature: Apart from EHD effects, strongly influences absorption/desorption and formation of reacted films. This is one of the most important parameters.

3. Out-of-Contact Times: The bulk of chemically reacted films are formed during the out-of-contact periods. Performance depends upon the relative rates of formation during these periods and removal when passing through the contact.

4. Slide-Roll Ratio: Affects surface temperature which, in turn, influences oil film thickness, physio-chemical reactions, λ ratio and stress distribution. Can have a marked affect on performance.

5. Contact Stress: Frequently made the main variable in laboratory testing and its direct influence on scuffing and pitting is well established. Changes that arise to the microstructure in the specimens must be similar to those encountered at working stresses, so this usually imposes an upper limit on stress. The influence on surface temperature can be marked.

6. Specimen Material: Essential to use same material, usually steel, as that of the full scale component. Heat treatment and finishing processes must be similar. Direction of lay is important but, often, specimens are ground at right angles so allowance must be made for this departure in the analysis.

7. Water Content: Has a direct bearing on rolling fatigue. Ideally test and full scale conditions should be of the same order. Frequently a water content is appreciably higher than would be imagined, especially when using synthetic oils.

8. Vibrations: These have an influence on both scuffing and pitting. From a vibratory survey of the full scale gearbox and test stand it is possible to operate at equivalent loads in "Back to Back" testing. Despite the amount of work done in this field it is not yet practicable to allow for differences in vibration patterns in the usual laboratory tests.

9. Filtration: The level of filtration can have a pronounced influence on rolling fatigue lives.

Test Machines

It is all too rare to find Engineers and Scientists working together in a multi-disciplinary Tribology laboratory and this tends to be reflected in the design of test machine that is available off-the-shelf. It is not the intention of this paper to criticise such machines as are available, sufficient to state that their limitations appear to be coming more generally recognised. The author and his colleagues have yet to find one that meets requirements and hence the need always to design our own machines, which in our case, range from simple cone-on-cylinder through progressively more sophisticated machines until reaching a back-to-back gearbox rig. Provided a test machine reproduces the required conditions reasonably well it can be used with confidence once a correlation has been established with full scale testing. A low inertia system is to be preferred so that minor changes in torque can be recorded. Based on the considerations listed above it will be clear that, in general, standard ball machines are unsuitable for gear simulation. In fact, results obtained from such machines can be very misleading.

Tests must terminate at the same stage of damage so it is useful to incorporate an automatic cut-off device in the machine. Pitting tests in point contact for example can be reliably be stopped at a predetermined vibration amplitude.

Surface Treatments

The bulk of standard evaluations will have been carried out with untreated surfaces and, quite often, the question arises as how to allow for the various forms of surface treatment without recourse to fresh testing. Treatments will alter surface texture and chemical composition of the surface layers and can influence both scuffing and pitting performance in a way that is not always obvious.

In general terms the treatments can help during running-in, in that they will enable the gears better to withstand rough handling but, this apart, some tend to act negatively. It is suspected that in such cases the treatment is providing competition and preventing the oil additives from performing their tasks effectively. Under steady state conditions it has repeatedly been found that, with few exceptions, bare steel/steel combinations out-perform treated surfaces. Proprietary phosphating treatments tend to be good but only provided care be exercised during the phosphating process.

For guidance purposes the indexes listed in Table 1 can be used. They were derived from disc machine scuffing tests using EN39B steel in combination with a 7½ cSt di-basic acid ester oil to specification ENG.RD.2487. In broad terms it has been found that pitting indexes are similar and, also, occasional testing with a different oil has produced results not too far removed from those predicted by applying the tabled indexes. The key word here is 'guidance' because not nearly enough work has been completed to imply anything stronger than that.

Treatment	Index
None	100
Black Oxide	69
MoS_2 - bonded	88
MoS_2 - sputtered	100
Proprietary phosphating (various)	127-158

Table I

The fact that pitting results rank in the same order as those for scuffing suggests that removal of the treatment with running need not be taken into account. Presumably this is because pitting lives tend to be strongly influenced by conditions ruling at the start of a test. The safest treatment, to avoid subsequent competition, is to cook the gears or other specimens in the oil to be used. It is found that a few hours at the tempering temperature of the steel will provide enhanced surface protection but to no great extent. On occasions it is difficult to run-in gears, when operating with a 5 cSt. oil without seizing. Pre-cooking in the oil has been found to help.

Heat Treatment

Variations in the microstructures have a relatively strong influence on pitting but less so on scuffing. Controversy still rages over whether to sub-zero temperature treat gears, and the position remains confused. Usually interest is centred on the influence of retained austenite on bending fatigue but the effect on both scuffing and pitting can be equally marked and, with respect to the former, in a surprising way.

Fundamental studies into the influence of austenite on scuffing confirm intuition that austenite, being relatively passive and appreciably more so than martensite, its presence in quantity will tend to lower scuffing performance. Repeatedly disc machine testing has given better scuffing results when high levels of retained austenite were present. The author's feelings are that in reducing the austenite level by sub-zero temperature treatment other changes in the microstructure are introduced and it is these that are responsible for the better results. This is a field that requires further detailed study with types of steel currently being used in helicopter transmissions.

Nickel content in a steel is generally found to reduce scuffing performance. EN31 steel has no nickel whereas EN39B has some 4%. Results of running disc machines tests with combinations of sub-zero temperature treated EN39B (retained austenite level approximately 14%), untreated EN39B (austenite level averaged 35%) and EN31 are given in Table 2. The oil used was END RD 2487. Times to incipient pitting are set out in Table 3 where the test conditions were similar to those for scuffing.

Disc 1	Disc 2	No. of Tests	Mean Failure Load, Lbs	Range Lbs.	Reference Index
A	A	2	1300	1300-1300	100
B	B	4	875	700-1000	67
C	C	4	1200	1000-1400	92
A	B	2	1050	1000-1100	81
B	A	4	1075	900-1300	83
A	C	2	650	600-700	50
C	A	3	800	700-900	62
B	C	2	750	600-900	58
C	B	2	750	700-800	58

A - Untreated EN31
B - Sub-zero treated EN39B
C - Untreated EN39B

Table 2 (SCUFFING)

Disc 1	Disc 2	Hours to failure
A	A	13
B	B	25
C	C	48
B	A	26
C	A	51

A - EN31 Untreated
B - EN39B Deep Frozen
C - EN39B Untreated

Table 3 (PITTING)

It will be noted from the above listed tables that pitting results are as to be expected and, clearly, sub-zero treated EN39B reduces performance. Results for scuffing are far from clear and, despite the introduction of a non-nickel containing steel as a reference, no obvious pattern emerged. The difference in performance between treated and non-treated EN39B is marked, in terms of 69 against 92. As before, these indexes can be used for guidance only because different operating conditions and oils are likely to influence results.

Correcting Results

The influences of the variables, as listed previously, on both scuffing and pitting performance have been studied in varying degrees of depth by many laboratories and results are available from the literature. The golden rule is that both mechanical and chemical aspects have to be taken into consideration. Usually a paper will relate to one of these aspects only, so extracting the information required is not all that straightforward. A further complication arises from the tendency of authors to draw generalised conclusions for which there is no justification. It really is necessary to ascertain the limitations of such conclusions. A general statement such as A is better than B can be most misleading unless the conditions under which this applies are set out precisely.

Two items not covered sufficiently well by the literature are discussed briefly below.

Direction of Lay: As a general guide, rolling across the direction of lay will increase minimum oil film thickness by 15% over the value for smooth surfaces. A 15% reduction applies when rolling along the direction of the lay. Obviously these values depend upon operating conditions but have been found useful when applying corrections.

Vibrations: It has been established that, under less than full film conditions, at certain frequencies both scuffing and pitting performance can fall off quite substantially. At higher frequencies performance is restored or even enhanced. This phenomenon has been studied in detail and satisfactory models developed to explain the performance variation. However, no satisfactory model has yet been evolved to allow for the influence of vibrations in the present context. The best that can be done in the laboratory is to run a few tests at different speeds and if corrections, to allow for the speed changes, do not bring the results into line with those of the main body then vibrations can be suspected and confirmed or otherwise from existing models.

Design Rules

For most purposes the rules laid down by such authorities as AGMA can be applied with confidence and will result in an efficient gear pair. In helicopter applications we are working at and beyond the limits for standard rules as well as, regretably, having to use an unsuitable oil, with a result that, for both pitting and scuffing, guidance has to be sought elsewhere. This usually means drawing on past experience as well as embarking upon a laboratory simulation programme.

Much work has been expended in attempts to formulate rules for predicting scuffing and pitting. With helicopter type steels and oils such attempts have, at the best, been only partially successful. However looking towards the future, if the day ever arrives when we have a single oil common to all helicopter transmissions the task of performance prediction will be simplified beyond all measure and practical rules will become realistic. What is badly needed is an updated version of Blok which, in slightly modified form from the original concept, has provided such a powerful tool for about 40 years when dealing with straight oils and moderate speeds. For a single oil and a small selection of steels such should be by no means impossible to achieve albeit on an empirical basis. The fact has to be faced that there are far too many variables in the composition of both oils and steels to enable a 'Building Block' approach to be adopted when modelling so one should seek little more than trends from such models.

Future Test Methods

On reflection there appears to be no reason to change what have become accepted test methods. Undoubtedly improved instrumentation will enable more information to be obtained from a set of tests. There is but one possibility that comes to mind that warrants further study. Accelerated tests for pitting that involve high loads tend to produce fatigue and other failures in the test machine. In general, test machine down-time often are unrealistically high. In certain case a basically similar oil, but with reduced load carrying capacity, can be formulated and used to keep test times down to an acceptable level. Obviously the corresponding loads applied are that amount lower. This approach has been used successfully in a wear generating exercise.

The more sophisticated a test machine, usually, the more expensive the test specimens and amounts of oil required. In practice, test costs rise exponentially with complexity of the test rig. A sensible philosophy seems to be to make the most of simple and cheap tests and reserve the more expensive ones for confirmation of preliminary results. Clearly at the end of the line there can be no substitute for full scale gearbox evaluation, but one must be completely confident of predictions before embarking upon such tests, on the grounds of cost, time and availability of components.

More use of 3-D graphics could usefully be made and probably will be in the future. By studying contacts throughout the torque range a good idea of localised contact stresses to be expected can readily be obtained.

Prediction Results

A list of prediction successes would serve little purpose, suffice to state that, with care, reliable estimates of full scale performance can consistently be made provided the conditions to be simulated can be ascertained reasonably closely. A few more unusual cases may, however, be of interest.

A long run of failures to pass a main gearbox through acceptance test without scuffing was causing considerably concern. Help was sought as a matter of extreme urgency to solve the problem within days. The time involved ruled out anything other than simple cone-on-cylinder tests of a type used for screening at the initial stages of developing a new oil. Oils to the same specification as that being used were evaluated. Results for the worst and best are shown in Fig. 5. It so happened that the worst was being used. A change to the best effected a complete cure. A cheap quick series of tests in this instance prevented further very substantial financial loss.

Because the selected oil performed so well on the above mentioned occasion it automatically assumed the role of a panacea and was recommended for applications for which it simply was not suited. A comprehensive study of its performance had been carried out in the laboratory, using disc machines and back-to-back testing. From such testing it was clear that the oil stood no chance of standing up to a 40% over-torque test in a main gearbox for which it had been recommended. The laboratory was able to predict a pitting failure at 27 hours. This actually arose at 27½ hours - the sort of thing that happens once only in a life time. As a consequence the laboratory's recommendation to change to a more suitable oil was accepted and the tests completed successfully.

One last example has been chosen as it occurred in the early days when we wished to satisfy ourselves that realistic full scale predictions could be made from laboratory simulation tes ing. A simple spiral bevel gearbox was monitored closely to record tooth flank temperatures, using an infra red technique, when operating with its standard oil. Instantaneous tooth contact ellipses were also measured, by using a lacquer/solvent method in which the solvent leached out the colour in the lacquer, throughout the torque range. Simulated disc machine tests were run using both the standard oil and a straight mineral base stock. This latter was chosen as being completely novel so that no previous experience could be brought to bear. The idea was to predict gearbox performance when using the mineral base stock. The result of the tests were encouraging in that in two tests pitting failure times clearly straddled the predicted value. Tooth flank temperatures were also recorded when running with the base stock oil and used for calculating the scuffing load, using Blok's concept. The predicted loads were in close agreement with experimental values.

In general, and bearing in mind that the actual conditions of contact to be simulated are rarely known, a reasonable time target for pitting would be $\pm$ 20%. This may appear a wide margin but scatter in rolling fatigue results under more ideal conditions than those imposed by gearbox testing is invariably large. A 40:1 ratio between longest and shortest lives in a reasonably sized sample of, say, 40 specimens is normal and hence the low slope of the Weibull line, usually not far removed from unity. Also the slope of an S/N curve is so shallow at gearbox operating stresses that a minor change in equivalent stress can result in very large differences in pitting lives. Scuffing predictions tend to be closer than this if they are of the type that can be expected if the load exceeds a certain level.

If performance of one oil or steel is being compared with that of another then predictions can be made with appropriate confidence levels quoted.

Comments on Figures

Fig. 1. Gives results of a small series of tests intended to demonstrate the influence that temperature imparts to different types of oil. Log stress is plotted against log cycles to pitting for both a medium and a powerful EP oil. At 50°C both oils share a single line whereas at 80°C the weaker oil's line drops below and the powerful oil rises above the 50°C line. These oils had the same base stock. The results illustrate the importance of surface temperature and it should be noted that ranking order of steel/oil combinations can vary with temperature.

Fig. 2. S/N curves for otherwise similar tests but at different slide/roll ratios are shown. These curves relate to high hardness steel specimens of approximately 750 VPN. The difference in performance mostly reflects the difference in λ ratios brought about through heating. This result is not, in fact, as obvious as it would appear because throughout the stress range the coefficient of traction for the lower slide-roll ratio case was higher than the other, appreciably so at some stresses. Traction forces also influence pitting performance.

Fig. 3. One aspect of the influence of vibrations on scuffing performance is illustrated here. Low frequency fluctuations in the applied load between rotating discs greatly reduced scuffing capacity at certain frequencies. These frequencies did not correspond to any regular pattern and resulted from the summing of two opposing mechanisms. Pitting lives broadly followed a similar pattern. This influence was found to apply under partial conditions of lubrication only and was not apparent when a full film was present.

Fig. 4. The relationship between B_{10} pitting lives and filter rating is shown here. The results were obtained from rolling element fatigue tests and are reasonably representative of helicopter gearbox conditions. It will be noted that substantial increase in life results from the introduction of fine filtration. In the present case the rate of improvement falls off below 3μm absolute filter rating.

Fig. 5. These traces, obtained from a Cone-on-Cylinder machine, compare friction behaviour between two oils to the same specification (ENG.RD.2497) which includes a load carrying requirement. The difference between the traces is marked. Smaller differences have been recorded between batches of the same oil.

Fig. 6. The better of the two oils illustrated in Fig. 5 was evaluated in a scuffing disc machine. With a reasonable gear oil, scatter in this form of testing is small. The very large scatter revealed with the ENG RD 2497 oil results from the 'Surface Temperature Distress Gap'. A test either passes through the gap, to produce high values of scuffing capacity, or fails to pass through to give the scuffing loads of the lower curve. The fact that in this small series of tests one of each type of failure occurred for each temperature is coincidence. Subsequent testing produced points on both curves and confirmed their general slope.

Fig. 7. Temperature measurements were monitored by thermocouples for oil sump temperature and spiral bevel gear tooth bulk temperature. An infra-red technique had been developed for measuring surface temperatures in the tooth bearing contacts, both mean and at discrete locations on the flank. The tooth bulk temperature thermocouple was located adjacent to the edge of the bearing pattern. In the graph, power and temperature are plotted against time in a staircase type test. On increasing the power a seizure/recovery event was encountered. The infra-red measurement shot up to an unknown, but high, temperature level whereas the two thermocouples gave no relevant indication. This example illustrates the care that is required in determining tooth surface temperatures - the tooth bulk value is not of direct interest. Temperature gradients are surprisingly high across a tooth surface.

Fig. 8. Electrical methods are sometimes used to monitor oil film thickness. The series of photographs shown illustrate the stages in building up an apparent full film (F) from full contact (A). In this instance a disc machine was left to run for 15 minutes without changing any of the conditions. In the main it is the resistance of chemical films forming on the surface that is being recorded. These films will tend to lower friction levels, and hence surface temperatures, so the actual oil film thickness will also be increasing marginally. When a full oil film is present complete dynamic decoupling between the discs becomes apparent. This did not, in fact, apply when level F was recorded. The chemical film can readily be removed, following which the system reverts to the situation as depicted in A.

Conclusions

Although much guesswork, or experience, has to be applied to full scale gearbox performance predictions from laboratory evaluations, they can be made to within acceptable confidence limits provided care is taken. A simulation laboratory has no alternative but to accumulate a comprehensive data bank and no opportunity should be missed to correlate results obtained from the various test machines with gearbox performance. At this stage but few hard and fast rules can be laid down and even these tend to fall down when working at the limits.

A mechanical approach in isolation is ineffective; it is necessary to consider other disciplines such as chemistry and metallurgy - a factor that is not always recognised in the design of evaluation machines.

Appendix

Composition of Steels

	C	Si	Mn	P	S	Cr	Mo	Ni
EN39B	0.14/0.18	0.1/0.35	0.25/0.55	0/0.015	0/0.012	1.0/1.4	0.2/0.3	3.8/4.3
EN31	0.9/1.2	0.1/0.35	0.3/0.75	0.05 max	0.05 max	1.0/1.6	-	-

Acknowledgement

The author thanks Westland Helicopters Limited for permission to publish this paper.

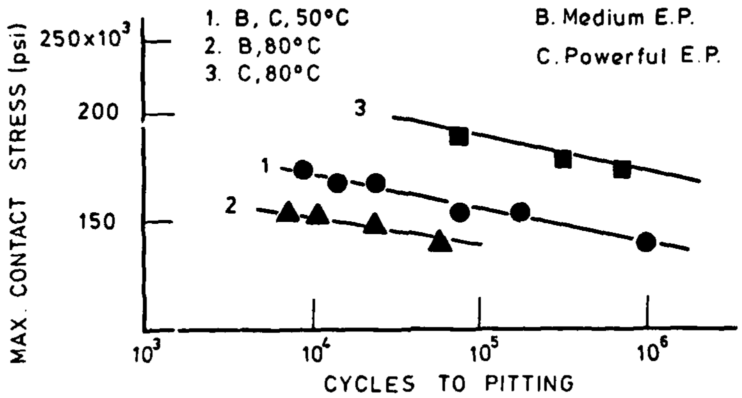

FIG 1. INFLUENCE OF TEMPERATURE ON PITTING WITH 2 OILS.

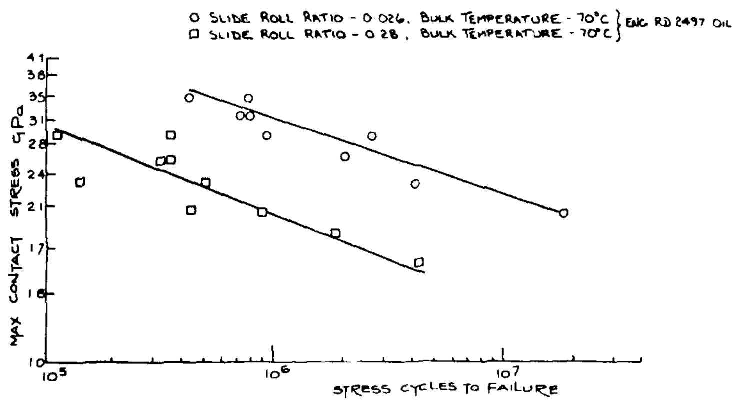

FIG 2. INFLUENCE OF SLIDE-ROLL RATIO ON PITTING.

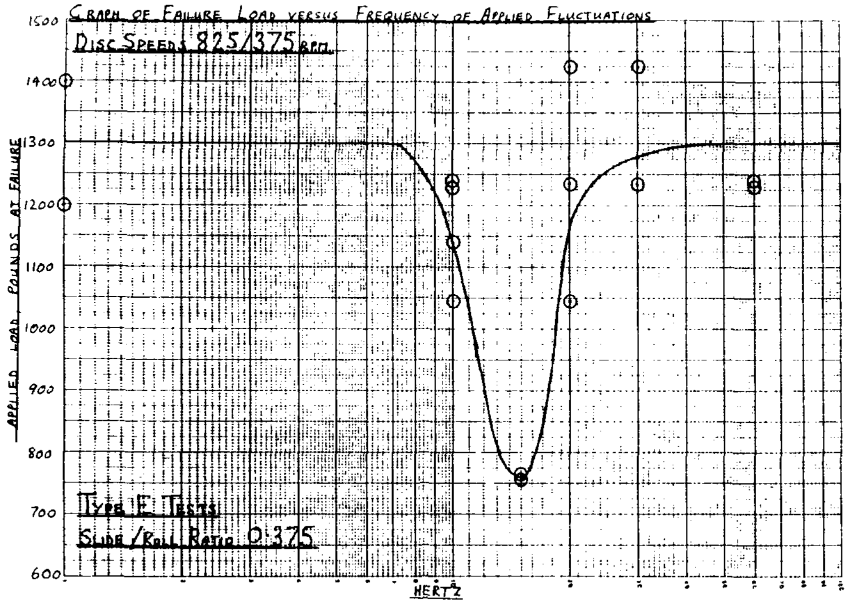

FIG 3. INFLUENCE OF VIBRATIONS ON SCUFFING.

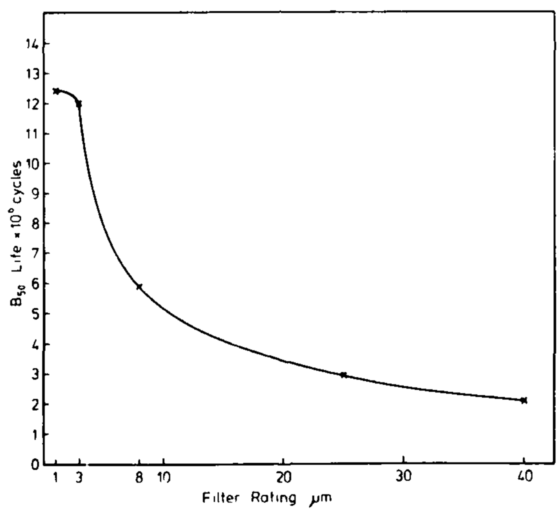

FIG 4. INFLUENCE OF FILTRATION ON PITTING.

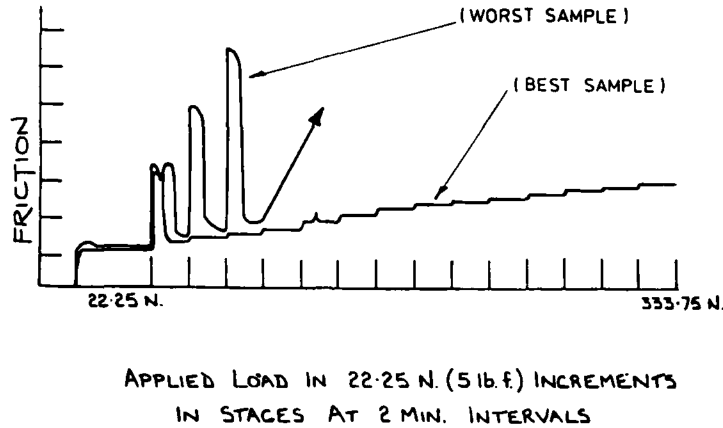

FIG 5. FRICTION TRACES FOR 2 OILS TO SAME SPECIFICATION.

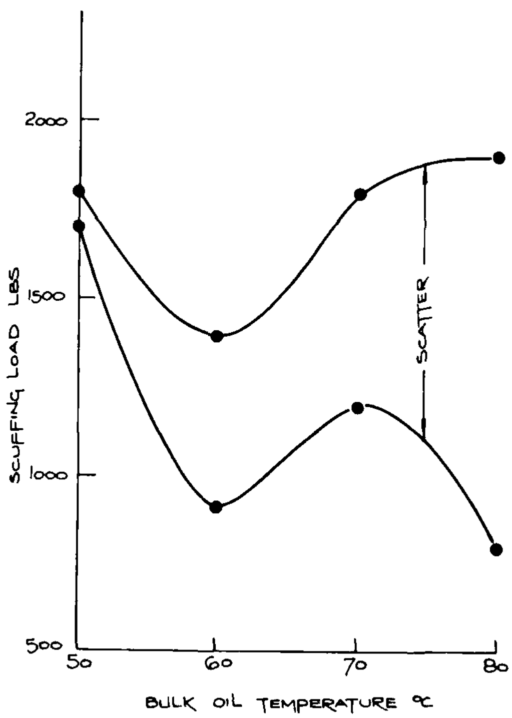

FIG 6. INFLUFNCE OF TEMPERATURE ON'
SCUFFING LOAD WITH ENG.RD. 2497 OIL.

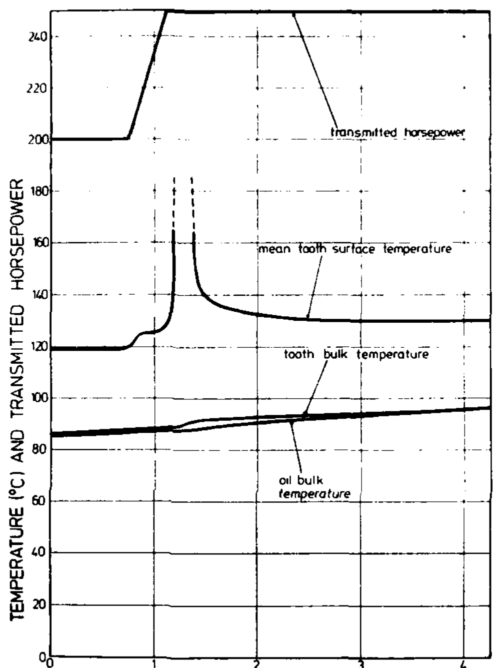

FIG. 20
TEMPERATURE TRACES DURING A SCUFFING/HEALING CYCLE

FIG 7. BULK OIL AND GEAR TOOTH

TEMPERATURES WITH INCREASING POWER.

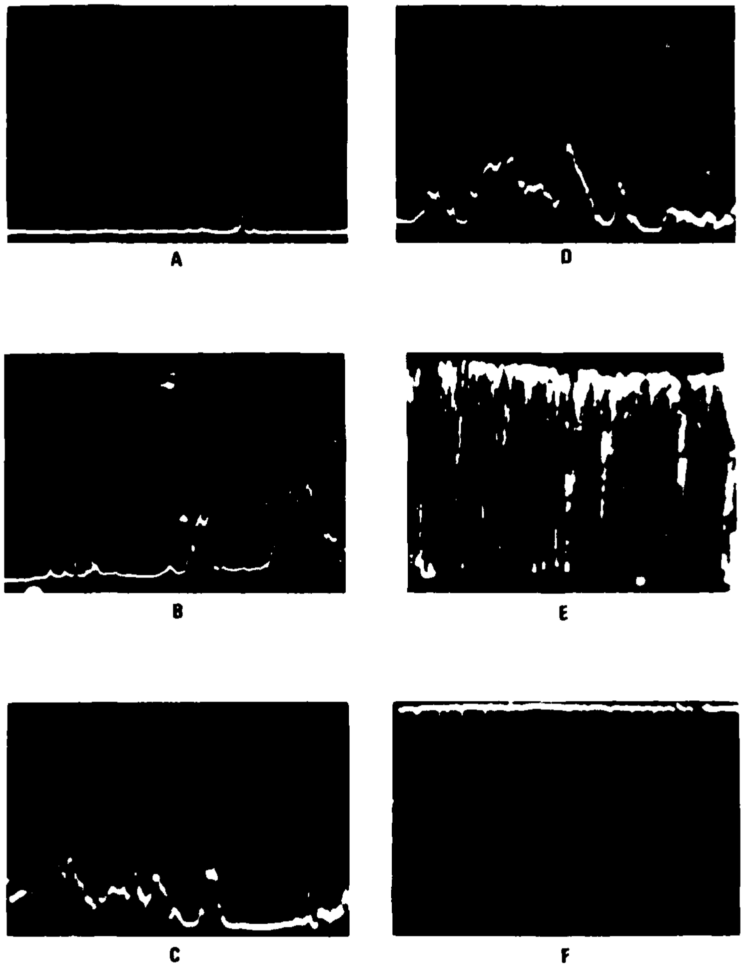

FIG 8. CRO TRACE OF VOLTAGE DISCHARGE, BUILD-UP TO FULL FILM.

EFFECTS OF UNFAVOURABLE ENVIRONMENTAL CONDITIONS ON THE SERVICE LIFE OF JET ENGINE AND HELICOPTER BEARINGS

by

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Aircraft bearings normally operate at high speeds and have to meet demanding reliability requirements. If conventional guidelines for their design are used, the resultant stresses are low. The stresses are particularly low if compared to those customarily applied in testing rolling fatigue. Testing at such low stress levels was in the past almost entirely relegated to extended performance testing.

It was only due to the inexplicably large differences between the calculated fatigue life (unfactored life) and the life actually achieved under normal aircraft-type loading and lubricating conditions that finally led to systematic fatigue life testing (1).

This testing was performed with angular contact ball bearings. The results showed that the standardized formula for the load-life relationship

$$L = (\frac{C}{P})^p$$

does not properly describe the life versus load relationship. Even at substantial stresses which are higher than many of those encountered in actual operation, the bearings did not fatigue provided that perfect lubrication prevailed. This proved that the S/N (Wöhler)-curve applies to antifriction bearings just it does to other dynamically loaded machine elements. Re-examining the bearings tested under moderate loads after testing verified this result (2). No structural changes could be found in the areas of the bearing exposed to the highest dynamic loading even after very long times of operation. Since structural changes caused by stressing such as plastic deformations could easily be detected as dark etching areas this signifies that no irreversible structural changes have occurred at the low loading level. If no changes take place in the material, there is no reason to expect the part to fail. The tests with angular contact ball bearings, from the results of which some important conclusions will be derived in the following, revealed that the maximum Hertzian contact pressure no longer leading to failure is close to p_0 = 2.5 GPa.

Endurance strength has been achieved at this stress level only if no unaccounted additional stresses are acting neither temporarily nor locally, such as hard particles which are allowed to enter the raceway. To account for these difficult-to-determine influences, we recommend the "practical endurance limit" (PEL) to be set at a lower stress level.

The experienced bearing user knows that fatigue failures can occur even at low rated contact pressures, if the bearing is exposed to contamination. The failure does not start below the raceway surface in such cases, at a depth where the highest stresses occur according to the theory which is the basis of the standardized calculation, but from the surface. Failure analysis of bearings operating in the field shows that fatigue failures almost always start from the surface.

In the following the difference between potential fatigue life and that calculated acc. to ISO standards, will be examined more closely. For bearing applications in the aircraft industry, the operating conditions as well as all other factors which have an influence on bearing performance such as manufacturing, inspection, assembly, and maintenance were defined on the basis of many years of practical experience. This led to a considerably surpassing of the unfactored life. When dimensioning new bearings, higher fatigue lives than those resulting from the standardized formula are assumed. Different bearing or engine manufacturers as well as regulatory agencies use different life factors. Often do they assume lower factors than those existing in reality thus taking over the role of a safety factor. The discrepancy is of course particularly striking if the existing contact pressures are below the endurance limit where no failure must be expected at all.

Excessive safety margins lead to unnecessarily heavy bearings thus aggravating the problems and risks caused by gravitational forces in high-speed or instationary operating conditions. They clearly increase the danger of skidding. Underestimating the actually achievable life prevents that the existing safety margin is properly taken advantage of. As a consequence the operational limits are reduced.

The Potential Fatigue Life

FAG angular contact ball bearings with the dimensions 25 x 52 x 15 mm, and z = 13 balls of 7.938 mm diameter and an inner ring raceway to ball ratio of 0.55, were tested on the rigs shown in fig. 1. Due to the test conditions, failure always started on the inner rings. The material used was:

SAE 52100, vacuum degassed,
austenitized at 1115 K
tempered at 450 K for 2 h.

The test was conducted on bearings from the same manufacturing batch at an inner ring speed of n = 12,000 min^{-1} . The cooled oil filtered through a fine-mesh filter ensured a sufficient separation of the surfaces at all load levels, so that after the tests no measurable wear could be found and that the inevitable deformations caused by smallest particles suspended in the oil (<5 x 10^{-3} mm) did not initiate any surface fatigue.

Figure 2 shows the relation of fatigue life to specific loading. The fatigue life is presented at the ratio of the test life to the rated life (ISO). As other tests have indicated these results may be transferred to other materials having comparable rolling fatigue strength as the steel SAE 52100 used in this test. The following classes of specimens were tested at various contact pressures:

Class A
The specimens which after the test showed a relatively deep circumferential groove due to plastic deformation were attributed to this class. In the zone of maximum pressure the bottom of this groove was 1.2 to 1.7 x 10^{-3} mm below the original surface.

Class B
The specimens showing smaller plastic deformation in the area of contact were attributed to this class.
Both classes contain the same number of specimens (n=10). The larger plastic deformation of class A bearings occurred despite the common heat treatment and is a result of the slight differences of the material properties; it is not due to different loading which is proved by the fact that the class of the largest plastic deformations even achieved the longest service life. The highest contact pressures in the center of the ball track have been reduced due to the plastic deformation.

Class C
These specimens were indented prior to the tests by means of a Rockwell tester (0.2 mm tip radius) with spherical plastic deformations 0.8 x 10^{-3} mm deep and 0.1mm in diameter located in the center of the ball track. The examination of the specimens after test showed that all indentations happened to be offset by a small, but uniform distance from the center of the ball track. This does not disturb the comparison between the various test loads.

The curves in fig. 2 demonstrate that the actual test results exceed the rated fatigue life even in the case of severe surface deformations caused by indentations.

The results admit important conclusions to be drawn as to the influence of plastic deformations, the influence of indentations as they occur when hard particles in the oil are cycled, and the influence of the stress level on the life of the bearings. These conclusions apply in the case of good EHD lubricating conditions.

The Influence of Plastic Deformation in the Raceways
This influence is shown by the different fatigue lifes of classes A and B bearings. Even small plastic deformations in the micrometer range have a great influence on the service life. The area of the highest contact pressure in the center of the ball to raceway contact is partly unloaded by the plastic deformations in the material. This produces a more uniform pressure distribution with a lower peak pressure in the center and has a positive influence on the fatigue life (Class A). The difference of plastic deformation increases with the load and so does the difference in fatigue life. At load levels encountered in the field, i.e. the lowest load level used in the tests, plastic deformations disappear together with the difference in fatigue life.

This means in reality that for comparative fatigue testing under high contact pressures which lead to plastic deformation there is always an unknown influence of the plastic deformation on the fatigue life, which cannot be determined accurately, and which opens the door to erroneous interpretations of the results. An accidentally or intentionally produced heat treatment can, for instance, conceal large differences in the qualit of the material which, however, may show up in actual operation at lower load le els.

The Influence of Surface Damage
As shown by the class C bearings with their defined indentations in the raceway the fatig . ife is substantially reduced in the presence of additional load disturbances in comparison with the classes A and B bearings. In such cases the fatigue failure starts exclusively at the damage area of the surface.

The curve C follows the theoretical fatigue life in accordance with the formula $L = (\frac{C}{P})^3$, except for a factor of 4. The curves A and B, however, which stand for the bearings where the failure started in the maximum stressed area below the surface deviate considerably from this formula.

Therefrom it is concluded that during the tests carried out decades ago to establish the exponents of the load/life formula, additional loading must have been present, probably caused by contamination of the lubricant as well as by metal-to-metal contact of the surfaces. This means that the fatigue life formula applies only for the bearing sizes and boundary conditions for which it was established.

The clear evidence of the considerable influence of indentations which have the same effect as solid particles entering the contact area should be discussed later on. It should be noted here, however, that indentations smaller than those used in the tests may explain any test results between the curves A and C.

The Influence of Loading

With clean lubricant and complete separation of the surfaces by an EHD lubricating film the rated fatigue life is surpassed by far, the more, the lower the loads are. Immediately below the lowest load level of the tests, where no evidence of plastic deformation could be found any more, the curves would asymptotically approach the horizontal Below this horizontal line no failure has to be expected: the bearings are below their endurance limit. With respect to the standardized fatigue life formula $L=(\frac{C}{P})^p$ this curve pattern means that the exponent p=3 at contact pressures above 3500 MPa increases gradually to infinite, if the contact pressure is reduced to values of about 2500 MPa. The standardized fatigue life formula does no longer accurately describe the load-life relationship under favorable lubricating conditions as they may exist in high-speed aircraft bearings. The following questions must be answered for real life conditions:

1. What conclusions must be drawn for the dimensioning of the bearings?
2. What is the influence of unfavorable lubricating conditions?
3. What conclusions must be drawn in the face of this new understanding of the load-life relationship when a higher loaded bearing application is to be evaluated in the light of the good experience gained with a very similar bearing application but at lower load levels? A similar question regarding the change in expected fatigue life occurs when a somewhat altered bearing support leads to slightly higher contact pressures within the bearing.

The Service Life under Favorable Lubricating Conditions

The new findings that bearings may reach an endurance range and that they do not fail as long as the stresses do not exceed the bearings' endurance strength must be taken into account. So has to be the fact that the actual operating conditions to which the bearings are exposed in practice always deviate more or less from the ideal , and that the load cycles and the actually acting contact pressures are seldomly known accurately. Because of this it seems prudent to use a "practical endurance limit" (PEL) for the time being because of these not accurately known influences and until further test data become available so that this endurance limit may be safe even when normal deviations from the design point which are not yet accounted for, occur, for example, during start-up and coasting of the bearings.

Ideal conditions are as follows:

- complete separation of the surfaces by a load carrying lubricating film, for example if $h_0/\Sigma R \geq 4$, whereas
  h_0 = rated lubricant film thickness
  ΣR = geometric sum of the average asperity height (AA)
- no detrimental contamination of the lubricant
- accurate knowledge of the amount and duration of loading, including all dynamic influences
- accurate assessment of the load distribution within the bearing, taking into account the actual rigidity of the surrounding parts.

The Practical Endurance Strength

Based on numerous tests with various internal designs or rolling bearings as well as with hardened and ground tooth flanks it may be concluded that rolling bearings are operating below their endurance limit as long as the static safety factor $f_s \geq 8$.

$$f_s = \frac{C_0}{P_0}$$

whereas C_0 is the static load rating of the bearing calculated on the basis of p_0 = 4000 MPa

and

P_0 is the static equivalent load. This value applies for the vacuum degassed through-hardening rolling bearing steel SAE 52100 as well as for materials exhibiting similar rolling fatigue strength such as M50 and T1. In special cases, especially in accurately controlled rig tests under laboratory conditions considerably better values are achieved The safety margin dormant in the "practical endurance limit" (PEL) as recommended should be maintained for actual operation until the results of further analyses of actual operating periods are available.

The Achievable Fatigue Life at Loads Above the Endurance Limit

Fatigue failures are to be expected if the loads exceed the endurance limit. In order to be able to determine the fatigue life for any load case the standardized load-life formula should be expanded by the load-depending factor f.

$L = a_1 \times a_2 \times a_3 \times f(\frac{C}{P})^p$

The validity of the factor f is limited to $f \geq 1$ (Fig. 3) or $f_s \geq 2.5$.

$$f = \frac{7.73}{(8-f_s)^{1.2}}$$

whereas $f_s = \frac{C_o}{P_o}$

Like the PEL the factor f also contains a safety margin. The factor may be increased occasionally if justified by positive operating experience.

The Achievable Life Under Boundary Lubricating Conditions

The complete separation of the surfaces cannot always be ensured in aircraft applications. In these cases, metal-to-metal contact occurs either temporarily or permanently. This leads to additional stresses due to friction and to heat generated by friction. The results are higher stresses in the area immediately below the raceway surface (3, 4) The material will then fail first at the location where its strength is exceeded most by the acting stresses. The area of critical stressing moves close to the raceway surface even in the cases of ideal contact geometry and surface finish if the metal-to-metal contact is intense. Without the metal-to-metal contact the distress would occur in lower layers of the material.

For the time being the potential life cannot be determined accurately for the boundary lubricating conditions as they occur in actual operation.

Each individual application has to be judged considering all experiences and test results available (5, 6).

Assuming good cleanliness of the lubricant and undisturbed rolling of the rolling elements, the following statements can be made:

1. All bearing designs which are suitable for high-speed applications exceed the rated life calculated in accordance with $L = (C/P)^p$ even under the conditions of severe metal-to-metal contact ($h_o/\Sigma R = 0.4$). According to (5), ball bearings made of the commonly used aircraft bearing materials M50 or T1 still reach a factor of 10 if all other operating conditions are favorable.

2. The fatigue life increases with increasing surface separation, especially with low contact pressures so that when the range of $h_o/\Sigma R = 4$ is approached, the fatigue life values within the limited life range previously predicted are achieved.

In order to completely describe the situation in the range of lubricating conditions, $0.4 \leq h_o/\Sigma R \leq 4$ a large amount of research remains to be done. Most of all, the influence of the contact pressures which normally exist in actual operation must be taken into account. Guiding values for the potential life may be obtained by reasonable interpolation between the two extremes.

Transfer of Rated and Empirical Values to Different Operating Conditions

Fig. 4 shows for class B of Fig. 2 the rated fatigue life in stress cycles of one point of the inner ring raceway and the actually achieved number of stress cycles under ideal conditions of lubrication and cleanliness. These curves are parallel only in the area of higher contact pressure. Assuming a small increase of the contact pressure it can be seen that in the area of 10^8 stress cycles the ratio between the increase of the actual fatigue life to the increase of the rated fatigue life is 2 while at 10^9 stress cycles the factor is 6.5 The ratio increases rapidly for even larger fatigue life values.

The following conclusions may be drawn for practical use:

Whenever, in actual operation, the L_{10} fatigue life exceeds 10^8 stress cycles it must be expected that the effect of changes of the contact pressure is not properly predicted by the standardized load-life formula. This must be taken into account be the designer.

Important: Load increases or increases of the individual rolling elements loads (such as by unfavorably supporting the bearing) may have a much more detrimental effect when compared with the actual fatigue life than predicted by the formula.

Local stress rises such as those caused by a faulty raceway geometry or by surface distress brought about by contamination may have the same negative effect.

Unexpectedly large fatigue life reductions as a consequence of increased loads may not always be accepted as the result of an unsufficient theory. This may be justified only in the range of high stress cycles. An unexpected fatigue life reduction often means that the higher loads have caused secondary stress increases produced for example by further bending of the shaft or deformations of the surrounding parts which were not taken into account in the original calculation.

The reduction of the fatigue life is often due to a deterioration of the cleanliness. The large influence of raceway indentations already shown in fig. 2 should be discussed in the following.

The Influence of Contamination

A detailed examination of the specimens of the classes A and B (fig. 2) showed that particles of a few micrometers in size had entered the contact areas despite of the extensive filtering of the oil (fig. 5) The 3×10^{-3} mm deep indentations caused by them had no influence on fatigue life. The examination of the failed bearings showed that structural distress occurred in greater distance from the raceway surface in the area of maximum stressing. In order to systematically investigate the effect of larger surface defects on fatigue life, more specimens were indented in the center of the raceway prior to the test as with the specimens of class C, by using very hard balls of various diameters (0.4, 1.0, 2.5 mm). Each specimen had four endentations equally spaced around the circumference. Test parameters were the ball diameter D, the diameter of the indentation d, and the depth of the indentation t (fig. 6).

Test specimens were again inner rings of the standard bearing FAG 7205B (25x52x15mm) of SAE 52100. The tests were conducted at a speed $n = 12000\ min^{-1}$ and under the same ideal operating conditions as mentioned previously. The rated maximum Hertzian contact pressure between inner ring raceway and ball was 2800 MPa. The evaluation of the test results showed the following:

The fatigue life of the specimens not indented exceeds the rated life considerably, as could be expected. The deeper the indentation and the higher the raised edge around the indentation the larger also is the reduction of the fatigue life. Indentations of same depth but caused by larger diameter balls resulted in a smaller reduction of fatigue life than those caused by smaller diameter balls (Fig. 6).

With a 0.02 mm deep indentation produced by a ball of 2.5 mm diameter the test life was clearly larger than the rated life determined in accordance with $L = (C/P)^p$. With an indentation of the same depth but caused by the smallest diameter ball (0.4 mm) the test result was $L_{test}/L_{rated} = 0.3$. A single indentation is sufficient to cause a fatigue failure. The most detrimental indentation determines the time to the start of the failure.

The influence of actual contamination particles on the fatigue life was determined by further tests and compared with the influence of ball indentations (see 1 to 5 in fig. 5). The inner rings of the bearings used for these tests were taken from the same production lot as the rings affected by the ball indentations. All test conditions were also the same.

The influence of the particles shown in Fig. 7 at large magnification along with the typical raceway damage they have caused correspond to 2 to 5 in Fig. 5. The test variation I in Fig. 6 does not contain any indentation, its raceways are undamaged.

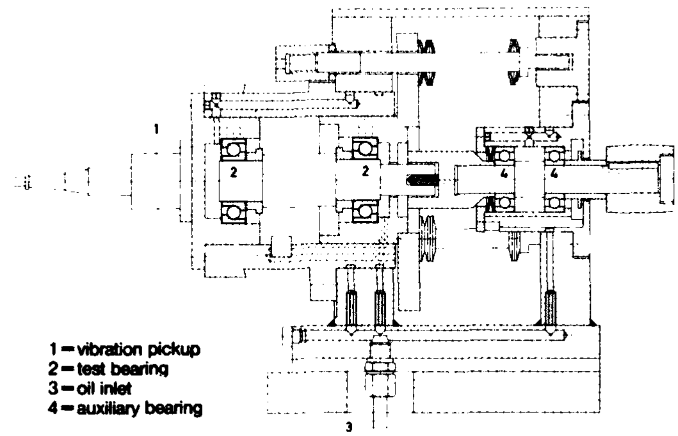
The scattering of the actual fatigue lives within a lot of 10 specimens is larger among the specimens indented by particles than with those indented by balls. This may be explained by the irregular shape and the scattering in the size of the particles, but also by the fact that the indentations were not always placed in the center of the raceway.

The tests described above indicate that damage caused to the rolling contact surfaces by ball indentations may reasonably be compared to damage caused by contamination particles, as far as their life-reducing effect is concerned (Fig. 6). The tests described below were therefore conducted using the well reproduceable indentations by means of a Rockwell tester with a tip diameter of .4 mm

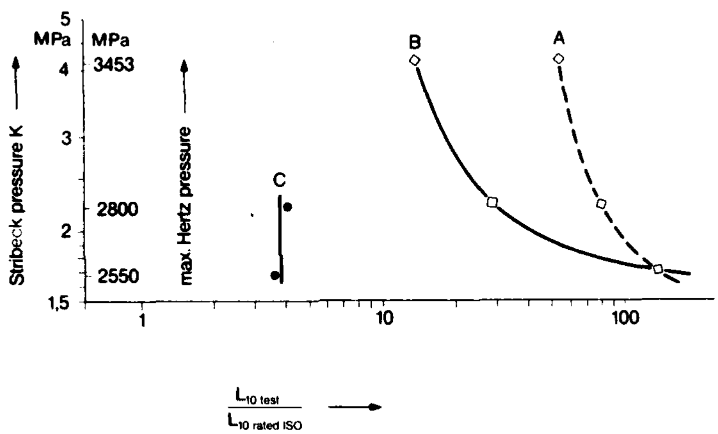
The Influence of the Bearing Size and Internal Design

Test specimens were the angular contact ball bearings FAG 7205B (25x52x15mm) and FAG 7312B (60x130x31mm) as well as tapered roller bearings FAG 518772A (29x50.3x14.732mm) All bearings were taken from the production line. They were axially loaded in the test (equivalent dynamic load 50% of the load rating acc. to ISO). The lubricating conditions were so favorable that unindented bearings exceeded by far the rated life values. Fig. 7 shows the relative running times ($L_{10\ damaged} / L_{10\ undamged}$) until failure of the bearings as a function of the pressure ellipse length. For a rough estimation of the influence of size and type on fatigue life, the assumption of a simple function of the length of the pressure ellipse seems to be sufficient. It is important to state that same-size indentations reduce the lives of small bearings with point contact much more than the lives of large bearings with point contact or of bearings with line contact. It becomes evident that the lubricant of small ball bearings has to be kept clean especially.

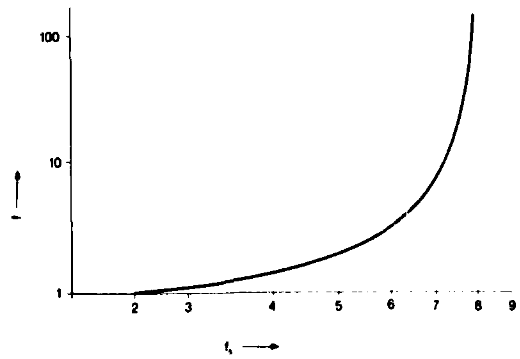
The presented influences can explain many unexpected bearing life values. The systematic application of the experience gained to the dimensioning and lubrication of the bearing arrangements will have a positive influence on reliability and economy of bearing arrangements. The failure-proof rolling bearing is no longer Utopia. This is especially true of jet engine and helipcopter bearings with contact pressures often in the range where no material fatigue has to be expected and favorable and clean lubrication conditions may exist. Jet engine and helicopter bearings benefit much from the fact that much more than in any other fields of technology, designers of aerospace bearings are aware of the necessity that lubrication, too, must be planned, carried out, and maintained diligently.



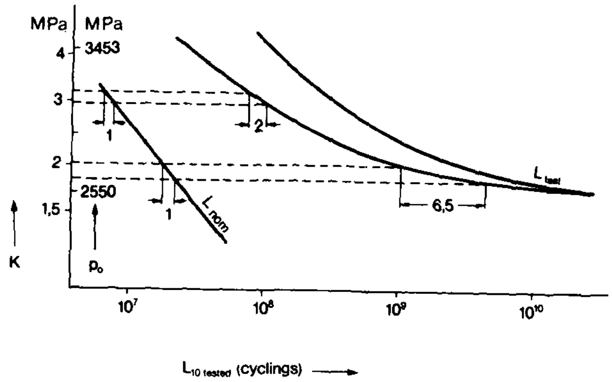
▲Fig. 1: FAG life test rig



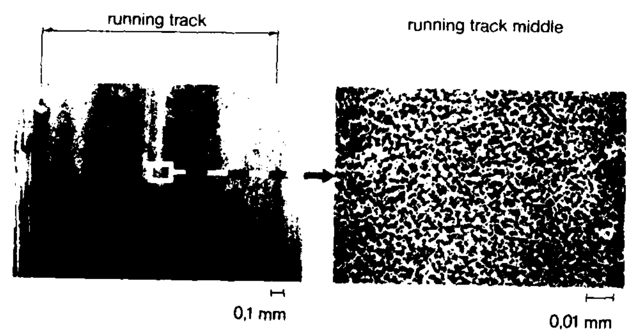
▲Fig. 2: Relation of fatigue life to specific loading



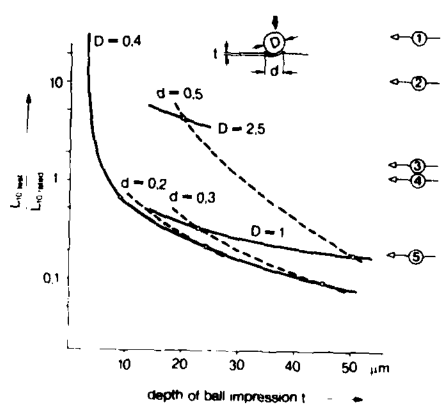
▲Fig. 3: Load-depending Factor f as a function of index of static stressing f_s



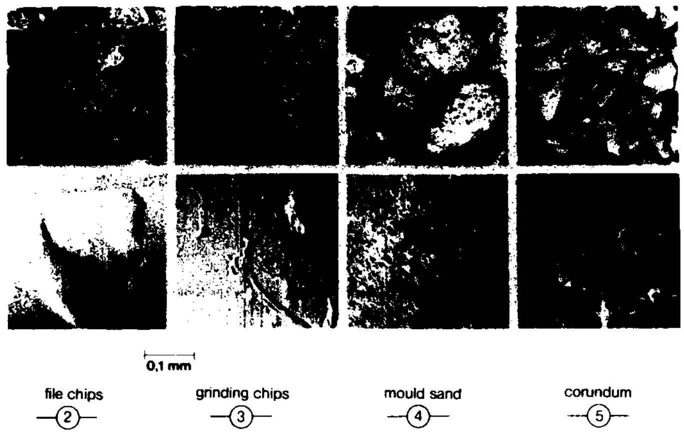
▲Fig. 4: Life under ideal lubrication and cleanliness conditions



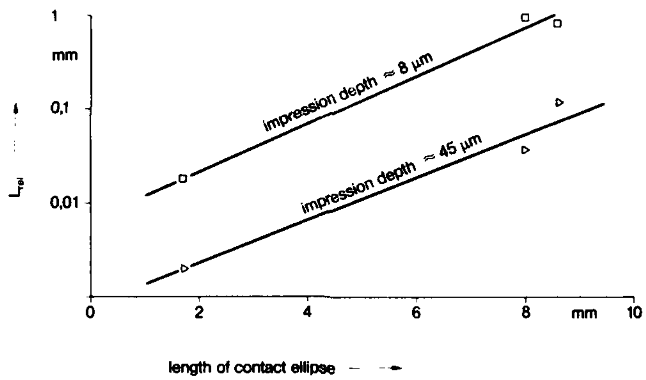
▲Fig. 5: Indentations in the rolling contact surfaces of a rolling bearing caused by finest, cycled contaminants.



◄Fig. 6: The influence of surface damage on the life of rolling bearings caused by ball indentations and actual contaminants



▲Fig. 7: Actual contaminants and their impressions in the rolling bearing raceway



▲Fig. 8: The influence of damage in the rolling contact surfaces as a function of depth of impression and length of the pressure ellipse

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Katalog 41 510 DB

AD-P005 055

AERO-ENGINE GEARS: MANUFACTURING CRACKS AND THEIR EFFECT ON OPERATION

by

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Summary

This publication describes the effect of cracks in gear splines. After black-oxidising of case-hardened gears, stress corrosion cracking may occur. This was determined from damage analysis of cracked gears ex flight engines and from sample testing. Besides design measures to reduce operational stresses, corrective action may be in the form of modified manufacturing procedures and improved crack inspection.

1. INTRODUCTION

Since some 20 years ago, damage analysis has been carried out at MTU on aero-engine components such as discs, blades/vanes, gears and others.

In the past, gear damage occurred repeatedly with similar damage patterns. Such damage occurred on the most varied of engine types (see diagram 1).

2. EXAMPLES OF DAMAGE

Three examples of damaged gears are described in detail below (see Fig. 1 - 12):

(a) Gear with cracked splines (see Fig. 1 - 3)
(b) Gear with failed bearing shaft (see Fig. 4 - 6)
(c) Fracture of pto-shaft resulting from cracked splines (see Fig. 7 - 12).

These three types of damage show the following common characteristics:

(a) The gears were all made of the material AMS 6260/6265
(b) The splines were case-hardened
(c) The gears were black-oxidised [1]
(d) The incipient cracks showed an intercrystalline, brittle fracture surface.

3. FAILURE ANALYSIS

Failure analysis was carried out on the fractured pto-shaft (see Fig. 7 - 8) to determine the cause of the fracture. Microscopic and metallographic analysis of the fracture surfaces is represented by Fig. 9 - 12.

The examination revealed the following:

(a) The gear fracture originated at a 1-2 mm deep incipient crack which occurred in the spline bottom. The spline surface showed distinct intercrystalline corrosion. The incipient crack showed a dark-grey deposit on the intercrystalline, brittle fracture surface. There was no evidence of a material defect. Microstructure and hardness of the case-hardened zone were satisfactory.

(b) The incipient crack had given rise to a fatigue fracture because of the operating stresses, which then caused the failure of the pto-shaft.

(c) Intercrystalline stress-corrosion cracking (SCC) which presumably occurred during chemical solution treatment in manufacture (e.g. nital-etching or black-oxidising) is suspected [2] . The production operations of the pto-shaft splines are listed in Fig. 13.

4. RING-SPECIMEN TESTS

Tests were carried out with specimens to reproduce the cracks in the splines on the pto-shaft. The tension test fixture and the specimen geometry can be seen in Fig. 14. The geometry of the splines was simulated in extremely simplified form with drilled, case-hardened steel rings. In order to produce a tensile stress in the ring holes while in the tensile testing fixture, the rings were slotted to permit tensile stress to be exerted on the specimen via disc springs.

The rings were then nital-etched and black-oxidised while under stress (see Appendix, page 10). These processes were carried out individually or in combination. The intercrystalline corrosion described above could be reproduced during nital-etching. It was also found that cracks occur in the holes during black-oxidising (see Fig. 15). Initial cracking occurred in the holes at a tensile stress of around 950 MPa. This value is clearly below the 0.2% proof stress of the case-hardened steel. As shown in Fig. 16, it is clearly a case of intercrystalline, brittle fracture [3]. These fracture surfaces have the same appearance as the incipient cracks on the gear wheel (pto-shaft).

5. TESTS ON TENSILE TEST SPECIMENS

A second series of tests were carried out to define the tensile stress in the specimen more precisely. For this purpose, round specimens with a circumferential notch were used. The geometry of this notch is derived from the geometry of the splines on the pto-shaft. The tension testing fixture and the specimen shape are illustrated in Fig. 17 and 18. The tensile stress in the specimen cross-section was varied between 0 and 1160 MPa.

Results:

— Intercrystalline cracks (see Fig. 19 and 20) occurred during black-oxidising of the specimens (dwell-time in the solution: 30 min). The cracks appeared in the bottom of the circumferential notch.

— After black-oxidising, the specimens were penetrant crack inspected. Cracks were found in the specimens under stresses of > 400 MPa. Next, the tensile specimens were loaded incrementally to rupture in a stress rupture test rig. Evaluation of the fracture surfaces revealed that the extent of the intercrystalline fracture surface is a function of the tensile stress during black-oxidising. This is represented in Diagram 2.

— Additionally, it was found that the specimens without cracks after black-oxidising (σ < 400 MPa) cracked during loading in the stress rupture test rig. The specimens were incrementally stressed in tensile to rupture at 150 °C in furnace atmosphere. The intercrystalline fracture surface region of the total fracture surface can be of various forms: isolated type, sickle type of ring type.

— Analysis of the fracture surfaces of the specimens which were black-oxidised under relatively low tensile stress, revealed characteristics of hydrogen embrittlement at the grain boundaries (see Fig. 21). The conclusion can be drawn from this that, under insufficient tensile stress, intercrystalline cracks can also develop from hydrogen embrittlement instead of SCC.

Alongside the tests described above, the effect of controlled shot peening on the rupture load of the notched circular specimens was investigated. Diagram 3 clearly shows that an increase in rupture load could be achieved by controlled shot peening of the notch.

6. CORRECTIVE MEASURES

From the failure analysis on the fractured pto-shaft and the related specimen tests to reproduce the stress corrosion cracking (SCC), corrective measures could be developed to reduce manufacturing cracks on the splines:

(a) During manufacture, it should be ensured that the splines are shot peened before black-oxidising. In addition, nital-etching should be kept to a minimum.

(b) As regards design, increased shaft thickness is recommended.

(c) Moreover, inspection of the splines can be improved with an additional eddy current crack inspection (see Fig. 22).

An improvement in quality is also possible for gear configurations similar to that of the pto-shaft. This reduces the danger of occurrence of such manufacturing cracks. The experience gained from damage analysis has shown us that, in extreme cases, a gear failure can cause the shut-down of an engine while in service.

In short, it can be said that the damage described here can be prevented or at least reduced by appropriate action during manufacure of the gears:

1. Case-hardened zones such as the splines, for example, should be shot peened before the final black-oxidising. This reduces residual stress which can arise during case-hardening. At the same time, the creation of compressive residual stress hinders hydrogen absorption.

2. Nital-etching which is used for inspection of the hardened teething should, where possible, not be used at all or used only once for critical zones such as splines, for example.

3. An effective incipient crack inspection process for the finished splines is the eddy current method.

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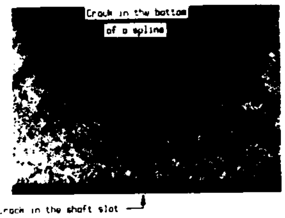
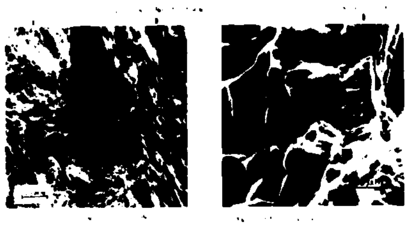
Gearwheel with splines

Fig. 1



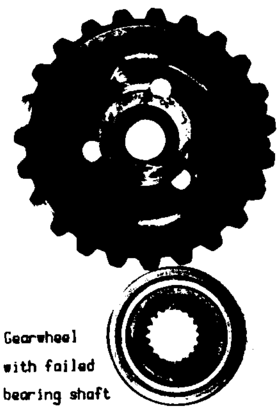
Cross-section through the splines

Fig. 2



Crack in the shaft slot

Fig. 3

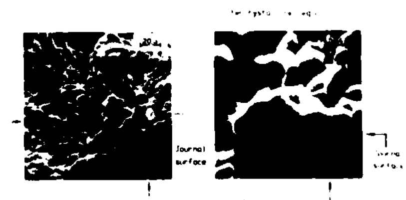


Gearwheel
with failed
bearing shaft

Fig. 4

FATIGUE CRACK PROPAGATION

Fig. 5



Intercrystalline region as starting-point of fatigue fracture

Fig. 6

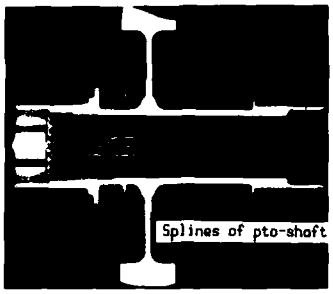


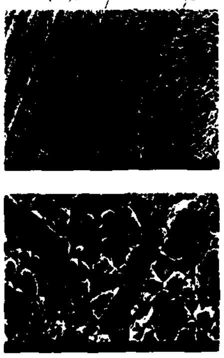
Fig. 7

Broken gear of the pto-shaft

Fig. 8

Pre-existing crack in the bottom of a spline
with adjoining fatigue fracture

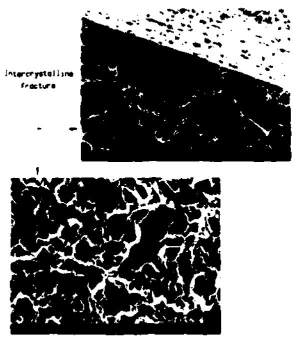
Fig. 9a



Intercrystalline corrosion in a spline

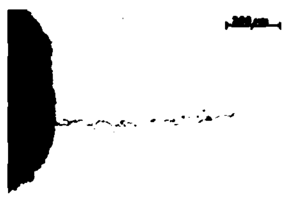
Fig. 10

Fig. 9b



Intercrystalline structure
of the pre-existing crack
at the bottom of a spline

Fig. 11



Intercrystalline course of crack in the bottom
of the spline

Fig. 12

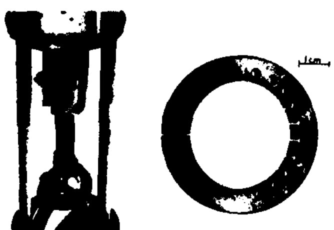
Production Operations (PTO shaft splines)

Shaft
- Broaching of splines
- Case – hardening
- Lapping of splines
- Nital etching

Shaft and **gearwheel** joined by EB welding

PTO shaft
- Nital etching
- Black – oxidizing
- Preservation

Fig. 13



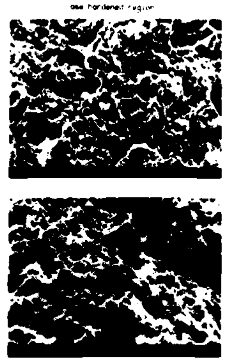
Test ring geometry

Fixture for tension testing of test rings

Fig. 14

Crack in the test ring

Fig. 15



Intercrystalline fracture surface of the test ring crack

Fig. 16

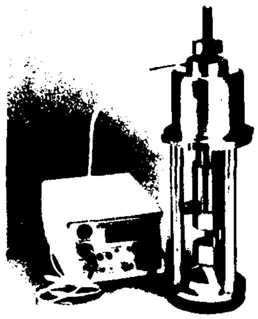


Fig. 17

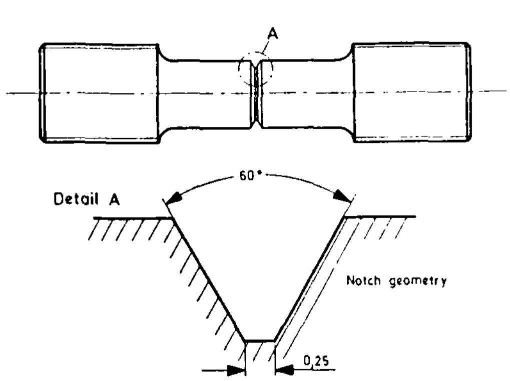
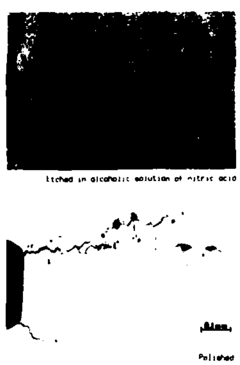


Fig. 18



Intercrystalline course of a crack at the circumferential notch in a tension specimen

Fig. 19

Fig. 20

Hydrogen induced intergranular stress corrosion cracking

Fig. 21

Eddy current test equipment

Fig. 22

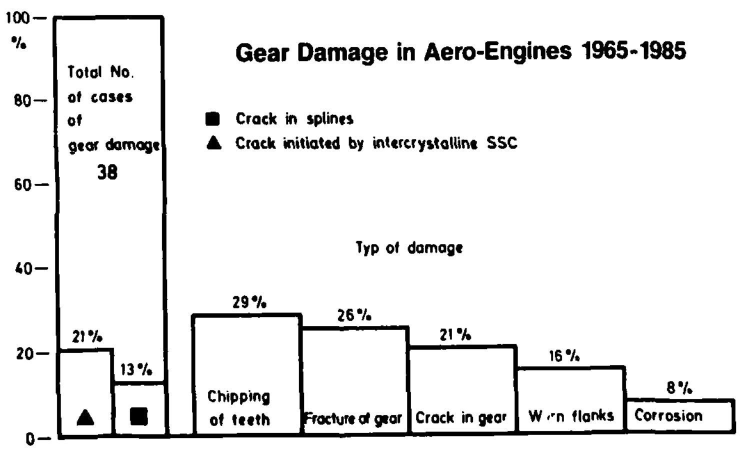


Diagram 1

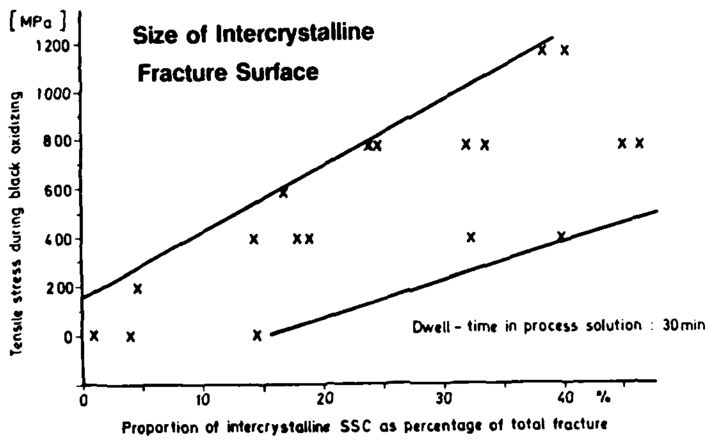


Diagram 2

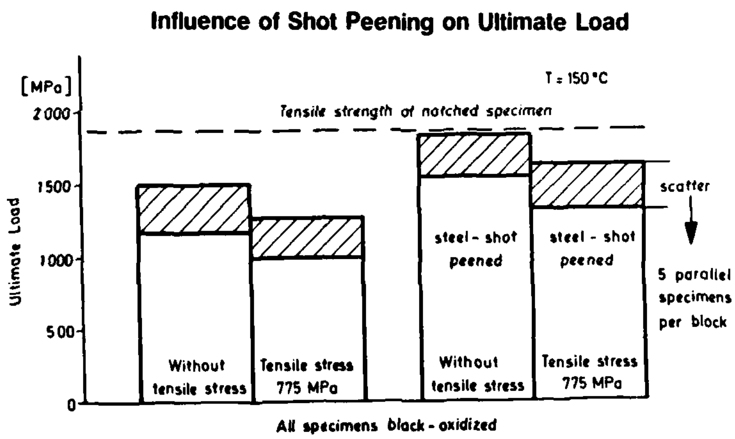


Diagram 3

Appendix

NITAL-ETCHING

- Perchloroethene – vapour degrease
- 30 – 60 sec. in alcoholic nitric acid (5% solution)
- Cold water rinse
- Dry with compressed air
- 20 – 30 sec. in alcoholic hydrochloric acid (10% solution)
- Cold water rinse
- Dry with compressed air
- Neutralize 40 – 60 sec. in alcoholic solution
- Cold water rinse
- Dry with compressed air

BLACK-OXIDISING

- Cleaning
- Cold water rinse
- 20 – 40 Min. at 140°C in $NaOH/NaNo_2$
- 5 – 10 min. cold water rinse
- Drying
- Immerse in preservation oil

TRIBOLOGY IN AIRCRAFT SYSTEMS -

BASIC PRINCIPLES AND APPLICATIONS

by

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Summary

The occupation with tribology in aviation is important for the future. To confirm this statement, at first the historical origin of the name tribology is outlined as a last step in an interdisciplinary approach, a linking concept, in the field of phenomena of friction of solids in relative motion. For a further confinement on tribology in aviation, the trends in this area for the next two decades are reviewed: reduction of fuel consumption, improvement of the ratio of specific output to weight, reduction of maintenance, further improvement of safety. Then the consequences from these trends are outlined with respect to research in tribology. The current status of research is sketched and some unsolved problems are discussed, e.g. interaction of material/lubricant in areas with mixed friction (reaction layers), interaction of material/lubricant in the hydrodynamic region concerning fatigue and concerning friction, friction and wear of synthetic bearings with and without lubrication, friction, wear and service time if solid lubricants are used, topographical features of surfaces. An outlook closes the paper with an appeal for early interdisciplinary government sponsored research to gain the wide spread of basic knowledge necessary for new and better solutions.

Introduction

Tribology in aviation is an important theme, it can be considered also as a statement, though last but not least it may be questionable for many people. That is why I want to start, like the scholastics, with the 'videtur quod non', the objection. It runs as follows: Why do we need tribology in aviation? Don't our aircrafts operate very well? And if there arise any problems, why not solve them as usual with the aid of experts, respectively. And there is one more objection: The notion that tribology is that an artificial product which actually does not mean anything new? Isn't it merely a new way of projecting something already well known?

These objections, which are raised from different sides, carry some considerable weight. Therefore, the main concern of the present work is to invalidate these objections and to show that understanding of and occupation with this field is the supposition for a timely and costly solution of problems in future for a large number of moving parts in aviation.

Why and what for is tribology?

Let us return to our objection that tribology is an artificial word, so to speak as formed in a retort. Everybody is in the right because the notion was first proposed by English linguists. But the essential point in question is the reason for creating this new word. Following the demands of the customers at that time, one will preceive that tribology is actually something very old, that this word represents not the beginning of setting up a new field of activity, but that it stands nearly at the end of a long thinking process. Describing it pictorially, this notion is practically the roof of a house which has been constructed in many details for a long time. For as long as several decades scientists of different disciplines who are engaged in the study of motion observe that there is a common and connecting phenomenon. This phenomenon is independent of size, shape and composition of the moving parts and also of surfaces, forces and ambient conditions. It is the phenomenon of friction of solid substances with motion relative to each other. Extending this idea over the entire range, one perceives that such extremely different events like downhill skiing, car-braking, extending the slats of a plane, a rotating a dentist's air-custioned drill, all belong together. The examples whose number can be enlarged arbitrarily, should show that a common phenomenon, namely the friction at variable relative motion, lies behind these manifestations for which correlations have nearly not been realized so far. The second basic aspect is the value of the coefficient of friction. This scale encloses at one end for example metal on metal friction with very high friction numbers and at the other end the exceptionally low friction in air- or vacuum-cushioned bearings (Fig. 1). Having got this knowledge, it was only a last step to create for all phenomena together an overall concept. This concept became, after doubtlessly not easy considerations, the Greek word "TRIBOS", that means friction. Tribology includes also the aspect of long enduring and unaltered good function of moving parts, i.e. investigation and minimization of damage belongs to it. Keywords are damage by fatigue, by fretting and abrasion (wear) (1,2). Also included is the branch of inspection and methods for premature diagnostics of damage. After all the question has to be put: How may that what happens in the technical

field of lubricated phenomena of motion be linked with the new area tribology? Now it is doubtless to realize, that tribology means more than lubrication. Just on the contrary one can state that lubrication is only a branch of the total field of tribology.

Now, with the word tribology, we have found a linking concept, but this does not automatically answer the question why and what for do we need this new concept for our work during the daily routine. The answer for this is: The interaction of all aspects which are supposition and accessoires for the optimal achievement of 'low friction' or low wear and service time, can be identified and taken into consideration during the application of tribology (Fig. 2). Besides the perception of the complexity due to various factors of influence, another fundamental aspect is the interdisciplinary nature of tribology. That means the fields of mechanical engineering, material science, oil chemistry, surface physics and chemistry. Finally we can say that future success in tribology will be decided essentially by the extent of the interdisciplinary functioning. Here it is to remember that other areas have already great success by interdisciplinary cooperation.

Tribology in aviation

Let us go back to the introduction and its objections. It is no point of discussion that airplanes do not only operate sufficiently good, but that they belong to the safest transportation system. It is also known that aircrafts embody a lot of in part extremely different moving parts which function predominantly well. Only some of the different major components like wings, body, engine, landing gear and steering mechanism or equipment (Fig. 3), which exhibit just as gears and roller bearings complex sliding mechanisms, may be recalled. Why here still tribology? Before starting to answer the question, it appears necessary to discuss the previous trends in aviation for the next two decades. In doing so we have to mention:

Reduction of fuel consumption

Improvement of the ratio of specific output to weight

Reduction of maintenance

Application of modern materials

Further improvement of safety

All of these aspects are extensive fields of activity with various partial aspects. At this point only trends and their reference to tribology should be shown.

The reduction of fuel consumption can be mastered above all by two measures. A medium term step is to increase at the engines the inlet temperature of the turbine, thereby improving the efficiency (Fig. 4) (3). A long termed step is the use of turbo-props with the latest propeller design as suggested by NASA (4).

In the first case, reference to tribology has to deal with the higher temperatures which now occur in the engine. One of the consequences will be also higher temperatures of the lubricant, with different types of change (Fig. 5), (5, 6, 7, 8). These oil-changes influence also the tribology. The example shows the load carrying ability between fresh oil and oil after use in high temperature engines (Fig. 6), (9, 10). Besides this, the additional diminution of viscosity will furthermore reduce the thickness of the lubricant film layer in the zone of contact. The consequences of material alterations, especially at medium and long operation times, cannot yet be overlooked but should be taken into consideration and investigated.

The new turbo-prop gears should allow to transfer extremely high power combined with a weight of this structural component as low as possible (11, 12). Hence will result high to very high stress of the teeth surfaces. How do oscillations, bending stress and vibrations in the new propeller system affect bearings and tooth surfaces is also an unsolved problem which requires consideration.

One should mention that on the area of improving the ratio of specific output to weight of aircrafts considerable efforts have been undertaken, especially during the last years (Fig. 7), (13). This resulted in the use of new materials and lighter dimensioned moving parts. This trend will be continued whenever possible. Therefore it leads automatically to the area of tribological questions and problems.

The reduction of maintenance in mechanical engineering is generally a task related to political economy. In this connection aviation has a special position due to its large requirements concerning quality and safety. Diminution of service requirements means for the area of tribology above all two different kinds: Firstly, no re-lubrication of bearings or oil change and secondly, no damaged moving parts which have to replaced. Progress has occurred in both sectors. One may only call to mind the use of so-called maintenance-free bearings, which partly comprise appropriate lubricants or are based on self-lubricating synthetics. Further improvements leading to lower wear and prolonged durability are herein necessary, too. Also oil change in aircraft turbines could be widely abandoned after experiments lasting for years. Instead of which, oil samples, especially from thermally heavy loaded aircraft engines, are inspected with respect to its changes in composition. From tribological points of view particularly the inspection of load carrying additives is of importance. It is necessary for the future to make efforts to standardize methods and evaluation procedures. Moreover, an aspect of growing importance is the influence of solid impurities in lubricants (Fig. 8), (14). Particularly the lifetime of roller bearings can be considerably improved by diminution of the number of such particles (15, 16, 17). Inspections concerning

this aspect, which were successful and have helped to reduce damage, should be developed and used furthermore (14). All the mentioned aspects indicate the same way, i.e. diminution of the requirements for service. Further activities in future concerning service and readiness for use are to be expected at this point, regarding the costs.

The theme of new materials in moving parts has to be considered under several points of view. One, as mentioned above, is to install mechanisms which need no maintenance and the other is to take novel or enhanced demands into account. These problems are the higher temperatures and the stronger mechanical strains which have to be expected in aircraft engines for the future. Both phenomena can appear simultaneously, (19). To achieve less or no maintenance composite materials become more and more important. Here, too, the 'component' oil will play an eminent role and will require fundamental tribological investigations. Additional factors can be the increasing rotational speeds in future as well as the stay period of solid particles during rolling. In this connection one has also to mention the lubricant. There may also be some necessity for modern lubricants independent of tribologic aspects. For example, some already mentioned reasons result in rising the temperature in the engine, which causes effects like damage of the oil, formation of deposits, corrosion or even self-ignition of the oil (Fig. 11). The influence of modern lubricants on tribology, that means the behaviour during sliding contact, can till now not be predicted by theory or calculation, but requires even now experimental investigation, which is not only expensive but also time consuming. That is why we would like to warn not to wait until the problem forces urgent measures but to initiate and execute advance research.

Connecting the theme safety in aviation with tribology, it is satisfying to say that there are no serious problems at present. Meanwhile the airplane is considered as one of the most safest vehicles. Heavy accidents during the last years can be traced back predominantly to human failure or violence. Changes in important moving parts, which may endanger safety, can be apprehended sufficiently early by means of various techniques for early recognition of damage. These techniques can be employed not only during the phase of flight, but also on the ground. However, a task for the future is to guarantee that the gears will work also after total loss of oil. This has to be fulfilled especially for a longer time for helicopters on tactical (military) missions to make it possible that they can land behind the own ranks. Another problem concerning the aspect safety is the use of new oils in hydraulic systems, which are less or not flammable (21, 22, 23). Tribological problems are wear and friction characteristics in different moving parts of the hydraulic system.

Trends and their consequences on tribology-research

Let us go back once more to the introductory question: Is, for the future, tribology absolutely necessary in aviation? Trends and their demands usually initiate new fields of research. The above mentioned trends in aviation, like the further improved ratio of performance to weight, the reduction of fuel consumption or for military purposes higher performance at the same fuel consumption as well as reduced maintenance requirements and large safety have, as shown already, also influence on tribology and its different sectors of research: These trends require early basic and applied tribological work to achieve the expected aim in aviation within the schedules for time and costs. An interdisciplinary approach is necessary, since due to the complexity of the interactions a straightforward improvement in one property can lead to mismatching with respect to other properties and consequently disturb the overall tribological system.

Some unsolved fundamental problems of aviation tribology ought to be mentioned in the following. The use of new materials or surfaces may possibly intensify the actual problems. These are:

- Interactions of material/lubricant in areas with mixed friction (reaction layers).
- Interactions of material/lubricant in the hydrodynamic region
  a. concerning fatigue
  b. concerning friction.
- Friction, wear and service time if solid lubricants are used.
- Topographical features of surfaces.

Interactions in the area of different types of friction represent a comprehensive field of activity. The consequences concerning material and lubricant, too, are multifarious and more empirically investigated than basically understood. Generally it is acknowledged that through chemical reaction a new layer has to be generated from the interaction of material and lubricant in order to separate safely the surfaces under high load (Fig. 11), (24, 25). Problems like how and how fast is such a reaction proceeding, what are the individual reaction steps, what is the effect of different materials and acidic impurities, are not completely solved, even in the case of phosphorus compounds (Fig. 12). These compounds are the main load carrying and antiwear additives in aviation oils. Furthermore, we like to show the different sensitivity of TCP as antiwear agent on different types of ball bearing steels (Fig. 13). The figures, too, reveal that not everything is definitely known. If other additives are used, similar problems and questions will arise. Moreover, the theme of the effect of aging products from bare oil or from additives is not sufficiently treated.

Not much is known about the interactions material/lubricant related to hydrodynamics if one considers the migration and the diffusion of lubricants which depend, for example, on material but also on oil components. There is also nearly nothing known about the effect of oil and oil components on the formation of microcracks and, therefore, on the fatigue of material starting from the surface. Little is known about the influence of oil or oil components on the friction at different kinds of material. A lot of basic problems are unsolved on the area of friction and wear of composite bearings, especially in the presence of oils. This concerns in the case of the airplanes mainly the instruments but also some parts of the steering gear and its moving parts. There is a wide variety of application of solid lubricants in aircrafts. Here, increasingly carbon fibrous materials became adopted also for high energy brakes. The knowledge about the adhesion depending on material and on the mechanism of the efficiency and its limits are, to a good extent, unsatisfactory. Also open questions are about the measurement and description of the surfaces and their alterations. This field, too, shows a lack for all kinds of moving parts.

Summary - Outlook

Tribology - the expression and the area is established since about twenty years. The reason of the founders was to create a deeper and broader understanding of friction and wear mechanism, that means the interaction of surfaces in relative motion. A part of this new science and technology is the field of lubrication.

In aviation, that means aircrafts, moving parts are used on a large scale and tribology is employed. Unfortunately, it is impossible to draw a happier picture with respect to our basic knowledge about 'tribology in aviation'. Experimental investigations in most of the relevant areas were only carried out if there were practical needs. We live, to a good extent, so to speak, still 'from hand to mouth'. Tribology as the interaction of physical and chemical phenomena is pursued in the most of the countries to a painful poor extent compared with other up-to-date disciplines. It is completely wrong and harmful to believe as hitherto that tribological problems are still solved by the manufacturers of oils and additives. For once it is no more their job to achieve on a broad basic the missing basic knowledge, and secondly the expenses have become prohibitive. There are also often no specialists available for interdisciplinary work. We can only gain a wide spread of basic knowledge if government sponsored research activities at universities and research institutes, similar to other modern fields of activity, are initiated.

Finally let us return to the introduction and its objection. It seems that there is no doubt any more that we need tribology in aviation. Its scientific support and technology will initiate new and better solutions for several areas and help to save costs and time.

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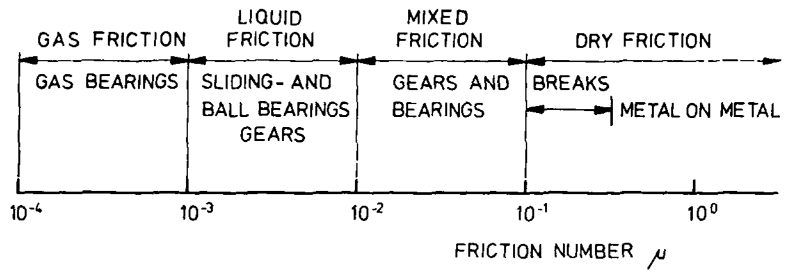

Fig. 1 Scale of friction number in the field of mechanical engineering

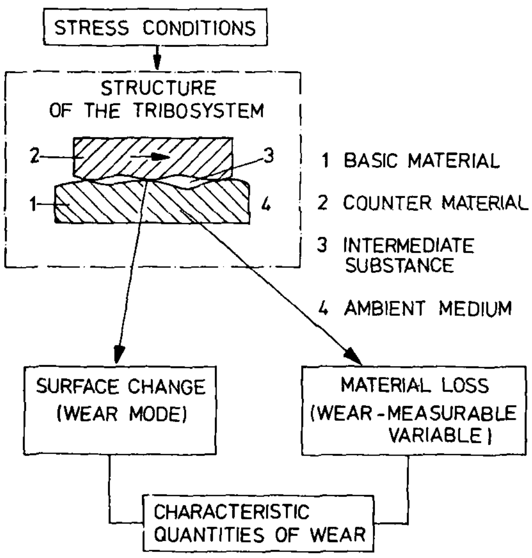

Fig. 2 Schematic of Tribology System used in the German specification DIN 50320

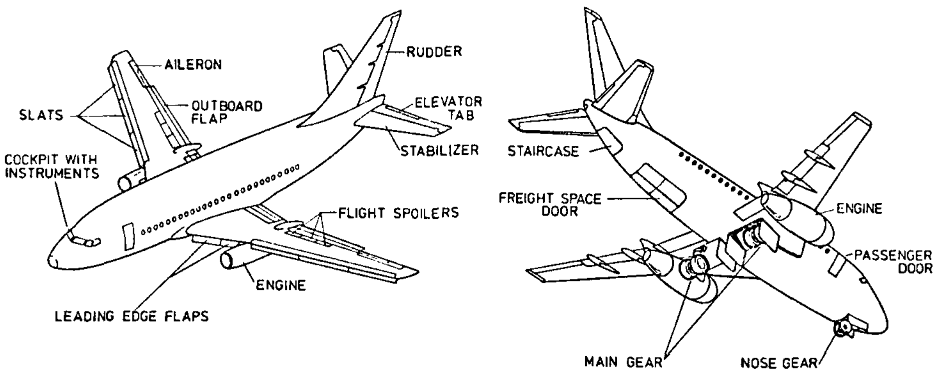

Fig. 3 Main sections of the aircraft and the respective moving elements

AIRCRAFT MAIN SECTIONS	MAIN COMPONENTS	MOVING ELEMENT
FUSELAGE	Doors, Staircases, Cargo Loading System, Internal Equipment	ALL TYPES OF BEARINGS, GEARS AND SLIDING ELEMENTS
WINGS	Spoilers, Aileron, Rotating Wings Leading Edge Devices	
EMPENAGE	Rudder, Elevator, Stabilizer, Gearboxes, Power Units	
ENGINES	Main Engines, Gear Boxes Auxiliary Power Units	
LANDING GEARS	Shock Strut, Wheels, Brake System, Retrack + Extent System	
COCKPIT	Gyro Horizon Indicator, All Types of Control Instruments, all Control Inputs for Flight Control and Landing Gears	
HYDRAULIC SYSTEM	Pumps, Power Units, Actuators	

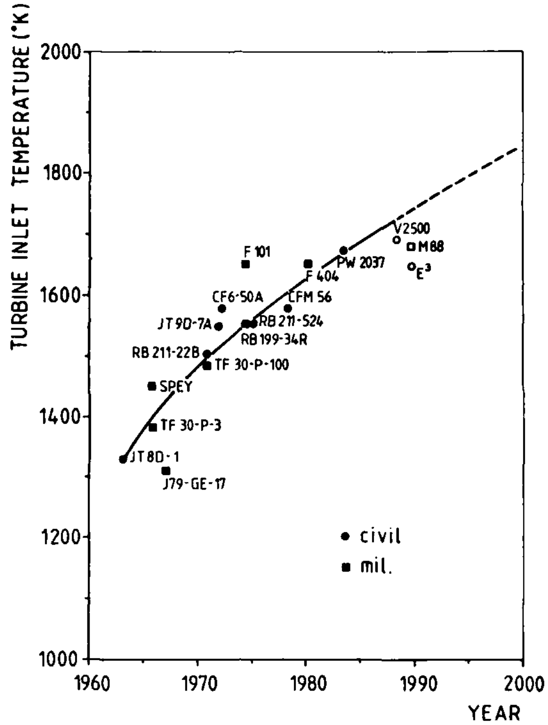

Fig. 4 Increasing trend of turbine inlet temperatures with time

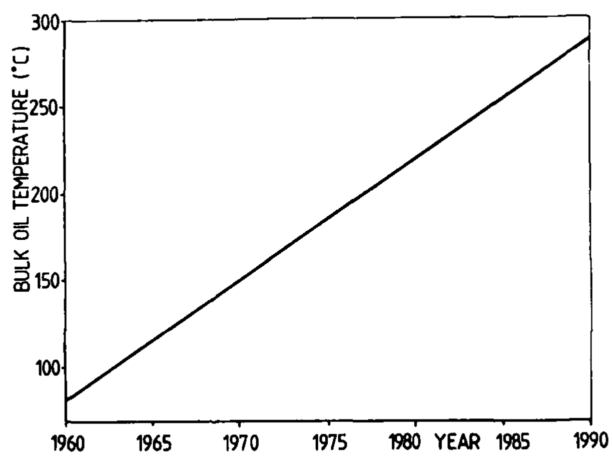

Fig. 5 Increasing trend of bulk oil temperature of aircraft engine with time

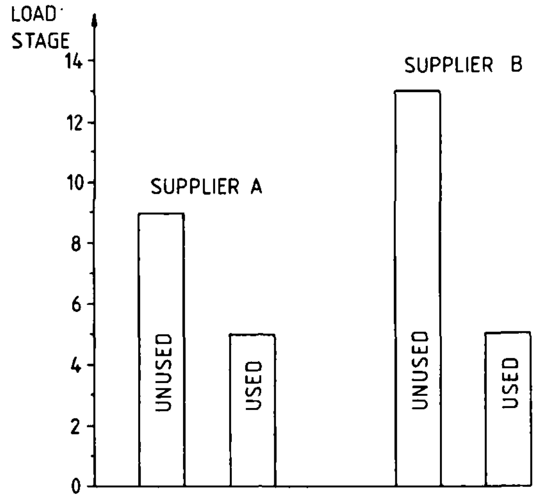

Fig. 6 Comparison of load carrying capacity of fresh and used oil samples using the FZG spur gear tester (DEng RD 2497 engine oils)

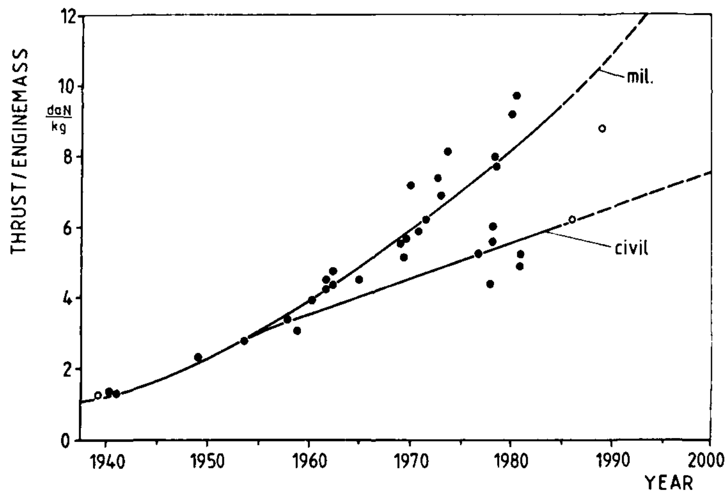

Fig. 7 Growth tendency of thrust to mass ratio of aircraft engine with time

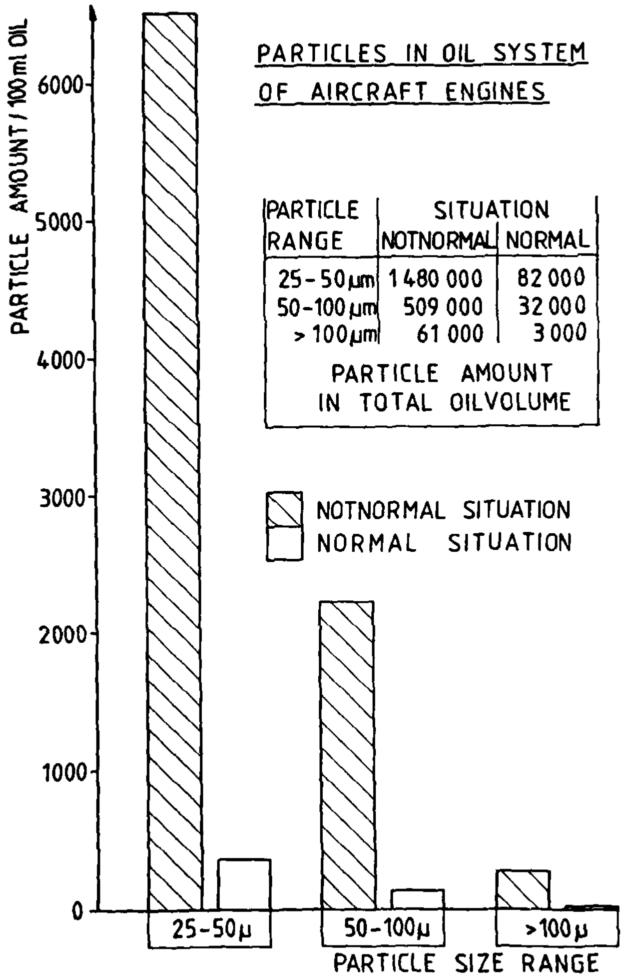

Fig. 8 Particle numbers in the oil systems of different aircraft engines

Fig. 9 SEM photograph of hard mineral particles obtained from a used engine oil

Fig. 10 Engine oil deposit developed in the housing of an aircraft engine bearing

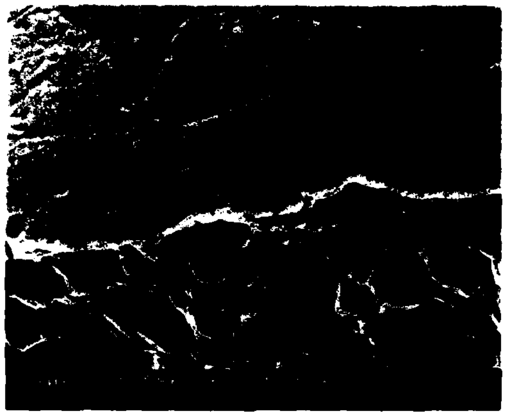

Fig. 11 Reaction layer developed by an engine oil on a metal surface

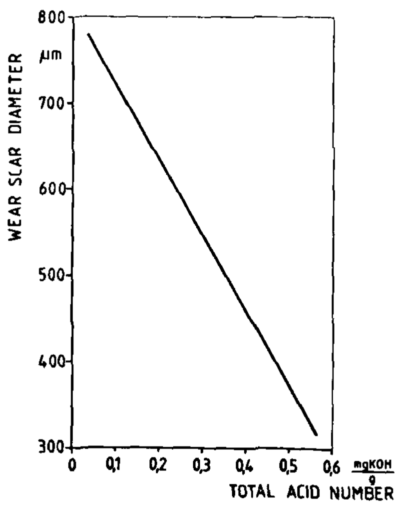

Fig. 12 Influence of acidic impurities, contained in commercial Tricresylphosphate samples on wear in the four ball wear tester

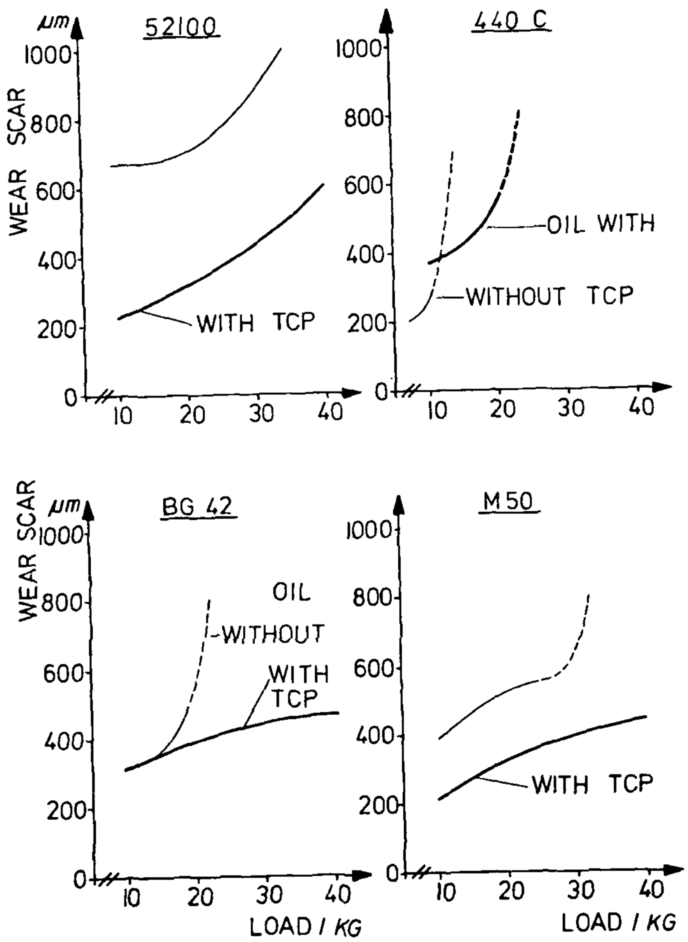

Fig. 13 Comparison of the effect of Tricresylphosphate on different steels in the four ball wear tester

AD-P005 057

DESIGN AND CALCULATION OF HIGH SPEED ENGINE BEARINGS

by

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SUMMARY

New aspects in design and calculation of high speed ball and roller bearings are discussed in as much as they affect bearing performance in the speed range of DxN = 3.0 x 10^6 min^{-1} and beyond. For roller bearings the guide flange to roller end configuration as well as roller excursion within the cage pocket are linked to the phenomena of skewing and skidding. The need for under-race-lubrication in small diameter engine bearings is derived from this and a special solution (integral scoop) is shown along with related performance data.

The influence of incorporating state of the art rolling elements made of silicon nitride on the design and calculation of high speed ball and roller bearings is shown taking into account hardness, density, Young's modulus and the Poisson ratio.

Increased speeds coupled with additional design features being incorporated (such as lubrication holes, grooves, slots) require that inner rings of improved fracture toughness be developed with existing materials of known properties thus supplementing the material development activities already under way.

Expressions used in this paper

a	mm	- roller end clearancee
d	mm	- diameter of rolling elements
D	mm	- bore diameter of the bearing
E	GPa	- modulus of elasticity (Young's modulus)
F	N	- force (load)
h	mm	- flange height
l	mm	- length of rollers
m	g	- mass
M	Nmm	- moment
nc	1/min	- cage speed (epicyclic)
n	1/min	- rotational speed of the roller about its own axis
N	1/min	- rotational speed of the inner ring
Po	GPa	- max. contact pressure (Hertzian stress)
T	mm	- pitch diameter
z		- number of rolling elements in a bearing
β	°	- skew angle of roller vs CL of bearing
γ	°	- opening angle of guide flanges
ν		- Poisson ratio
ρ	g/cm^3	- density

INTRODUCTION

It can be observed that the development towards higher density of energy transformation, which started with the earliest power generating machines, continues unrestricted even today. Since the focus of public attention has recently been directed to the latest developments in microelectronics and information technologies the continued development towards higher power density is more and more overlooked.

Naturally the trend towards higher power density is particularly strong in all airborn equipment and there again stronger in the main propulsion engines.

Generally speaking, the power density can be increased by increasing the three constituents entering the formula for energy transformation: speed, load and temperature. The characteristic temperature for aero engines, the turbine inlet temperature has increased dramatically during the last 2 decades based on the results of technical developments taking place in the area of turbine blade materials and designs. This is reducing the size of the engine core for a given power output. The higher turbine inlet temperature improves the thermodynamics of the engine. The mechanically moving parts, however , are not directly affected by higher turbine inlet temperatures, they are determined instead by the temperature limitations of the lubricants. There is little progress in this area since the introduction of diester based synthetic lubricants. Because of this, the trend towards higher power density concentrates , as far as mechanically moving parts are concerned, on the other two constituents of the power transformation formula, speeds and loads. The parts most affected by this trend are gears and bearings.

The subject of this paper, the bearings, are thus fully affected by ever increasing speed requirements. Loading, the other influencing factor, is not increasing by a similar degree, however, at least in those engine positions which are exposed to the highest speeds. The main reasons are the high reliability requirements of all airborn machinery and the fact that the size limitations are determined by gasdynamics thus allowing ample space for more than adequate sizing of the bearings.

In mainshaft applications of turbo engines with their finely balanced rotors there is no high demand on the ability of the bearings to carry radial loads except as a stand-by feature to contain the imbalance forces that may develop due to loss of blades or where gear forces are to be transmitted such as those generated by a radial drive train serving as power supply for the engine accessories.

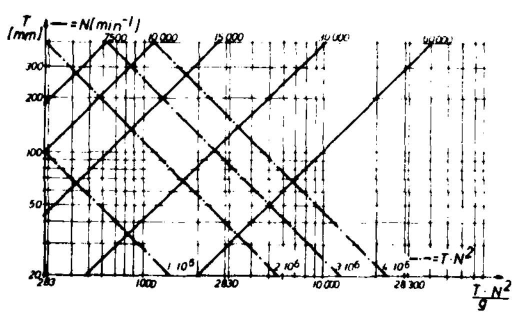
The characteristic feature determining the design of the cylindrical roller bearing is thus the speed not the load and will be discussed in the following. Because of the substantial thrust loads acting on the mainshaft ball bearings, even without imbalance due to blade loss, the load acting on the ball thrust bearings is an important feature besides their speed and will be discussed separately below in conjunction with the use of low density rolling elements.

EVALUATION OF BEARING SPEED

The traditional and most common measure of the speed of a bearing is its number of revolutions per minute N .This also being the speed of machinery it is directly related to the function of the bearing to transmit loads between parts rotating relativ to each other. As far as the stress and strain is concerned to which the bearing parts are subjected there are two measures of speed which more adequately characterize the performance of the bearing:

TN - The circumferential speed of the bearing which is characteristic of the sliding velocity of the bearing parts that are in rubbing contact with each other (cage to inner ring, rolling element to cage pocket or guide flange and so on). Customarily the circumferential speed is given as DxN [mm/min], which is the circumferential speed of the bearing measured at the bore diameter but devided by π. A similar measure TxN [mm/min] is based on the pitch diameter T of the bearing. When T is not directly known it may be substituted by the average of the outer and inner diameters. For high speed bearings TxN = 1.15 to 1.3 DxN depending on the relative size of the rolling elements. In the following TxN will be used as a measure of circumferential speed.

TN^2 - The centrifugal acceleration to which the rolling elements, the lubricant but also the cage and the rotating ring of the bearing are subjected may be characterized by TxN^2 [mm/min^2]. Here again the constant factor π is omitted as with the circumferential speed.



The nomograph from Fig.1 shows the circumferential speed TxN as well as the radial acceleration TxN^2 for combinations of T and N used in high speed bearings. The acceleration axis is graded in multiples of the gravitation constant g [m/sec^2].

FIG. 1 NOMOGRAPH FOR TxN AND TxN^2

HIGH SPEED CYLINDRICAL ROLLER BEARINGS

The main reasons for using cylindrical roller bearings in high speed applications are their ability to accommodate axial displacements without any build-up of a resistance force and their high load carrying capability in radial direction, both coupled with a low coefficient of friction, i. e. low power loss and heat generation. Ironically the high load carrying capability is not being utilized except in the unlikely event of a larger rotor imbalance caused by loss of blades. The duration of such an emergency, if it happens at all, is limited. This means that the designer of the high speed cylindrical roller bearing should provide for the ability of the bearing to contain large imbalance loads for short periods of time, but should concentrate on designing for low power loss avoiding and reducing friction whereever possible. In the following an attempt is being made to summarize the major influences on the power loss of high speed cylindrical roller bearings in order to visualize for the designer what otherwise may remain hidden within complex computer analyis such as the excellent TRIBO developed in conjunction with the parametric study on cylindrical roller bearings reported in [1].

ROLLER AND CAGE GUIDANCE - DRIVING VERSUS BREAKING EFFORT

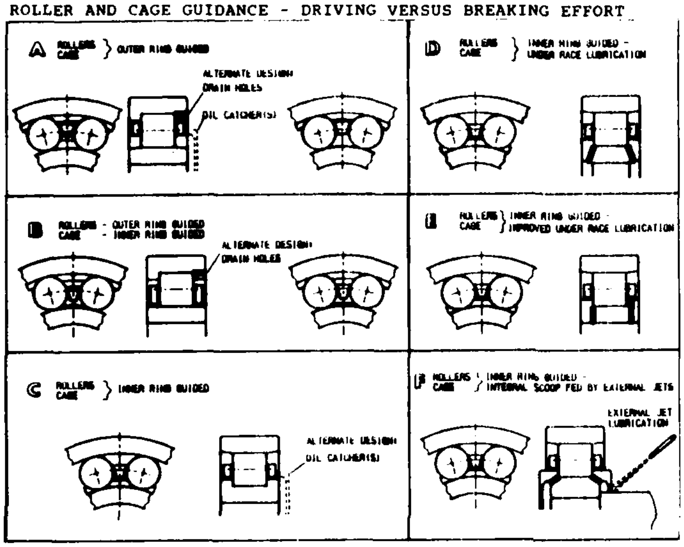


Fig 2 ROLLER AND CAGE GUIDANCE

The picture shows the basic variations A, B and C of guiding the rollers and cage. D, E and F are variations of the basic configuration C with respect to the supply of lubricant (Fig. 2).

Disregarding the driving effort provided by the (small) external radial load or by preloading the bearing internally by use of out-of-round races the following driving and breaking forces may be identified within the cylindrical roller bearing:

- rolling resistance of the rollers
- sliding friction between roller end face and guide flange
- sliding friction between cage and the ring surface locating the cage
- sliding friction between rollers and cage
- exchange of impulse between lubricant on one side and cage and rollers on the other side.

Each of these influences may provide a driving or a breaking torque on the cage and/or rollers and thereby support or dissipate the tractive effort produced by the external and/or internal radial load. Whether the influence is driving or breaking must be decided in each individual case by determining whether the respective influence tends to retard or to accelerate the rollers or cage.

The cylindrical roller bearing design A with roller and cage guidance by the stationary outer ring provides for good roller stability due to the large area of contact between roller end face and guide flange, an oil reservoir for start-up if the axis of the bearing is horizontal, self centering of the cage, easy mounting and good access for the lubricant (external jets). Unfortunately all the influences mentioned above are providing breaking torques thus counteracting the driving torque provided by the radial load. Since the friction torques increase with speed this design is limited to relatively moderate speeds unless the radial loads are substantial and always present. A particular problem with this design in high speed application is caused by the quantity of oil trapped between adjacent rollers, the outer ring raceway and its guide flanges by the centrifugal force acting on each particle of the lubricant. This quantity of lubricant is travelling at cage speed. Even with ample supply of lubricant to the bearing the trapped quantity may heat up to the point where its viscosity is not sufficient to prevent metal to metal contact between the guide flanges and roller ends. This problem may be alleviated by introducing drain holes to facilitate the exchange of lubricant.

The balance of all driving and breaking torques determines at speeds below the cage whirl speed the amount of speed loss of the cage (skidding) when compared to the epicyclic speed that may be computed assuming that the inner and outer raceways as well as the rollers are meshing gears. Aside from the cage skidding one should, however, also consider the skidding of the individual roller: The kinetic energy stored in the roller during its passage through the loaded zone of the bearing is more or less being dissipated while it travels through the unloaded zone. The moment the roller reenters the loaded zone the kinetic eneregy must be restored instantaneously providing for high shear stresses between roller circumference and inner and outer ring raceway.

In design B the friction between cage bore and inner ring raceway provides for a driving torque and reduced cage skidding but necessitates balancing of the cage and attention to proper lubrication particularly of the journal bearing surfaces made up by the inner ring raceway and cage bore.

In design C all the influences mentioned above - except the rolling friction of the rollers - provide driving torque, so that the demand on the driving torque provided by radial loading becomes a minimum. For higher speeds it is not possible to transfer lubricant to the inner ring raceway and guide flanges, however, because of the high centrifugal load to which each element of lubricant is subjected once it has assumed cage speed.

For very high speeds it becomes necessary to feed the lubricant from below the raceway of the inner ring. Design D and E show examples which are typical for those applications where the lubricant may be introduced through the hollow shaft. These designs also facilitate separating by design the small quantity of lubricant needed to properly lubricate all rubbing and rolling surfaces from the larger quantity of lubricant needed to remove the excess heat from the inner ring (under-race-cooling).

In some engines, particularly smaller ones, where feeding lubricant through the hollow shaft is not possible or impractical the design F is being used with good results by combining the advantages of external jets with those of under-race-lubrication.

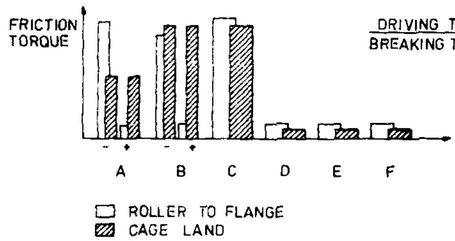


Fig. 3 FRICTION TORQUE OF RUBBING SURFACES

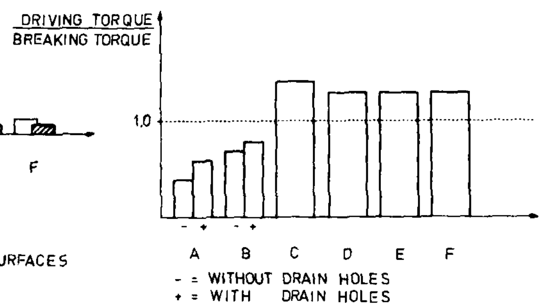


Fig 4 DRIVING VS BREAKING EFFORT

Fig. 3 summarizes the friction torques of all rubbing surfaces irrespective of their driving or breaking action.

Fig. 4 rates the various designs qualitatively as to their driving versus breaking effort.

Fig.5 shows the calculated cage speed as a portion of its epicyclic speed versus an external stationary radial load for a bearing having a pitch diameter T = 59 mm and square rollers with the dimension d = l = 5 mm. For a speed of N = 51640 1/min or TxN = 3.0 x 10^6 mm/min. The letter z indicates the number of rollers. The diagramm is in good agreement with experimental data gained by use of the FAG high speed test rig. The great influence of the roller mass is demonstrated by Fig. 6 which combines the critical speed for the various numbers of rollers of Fig. 5 in the case of outer ring guidance.

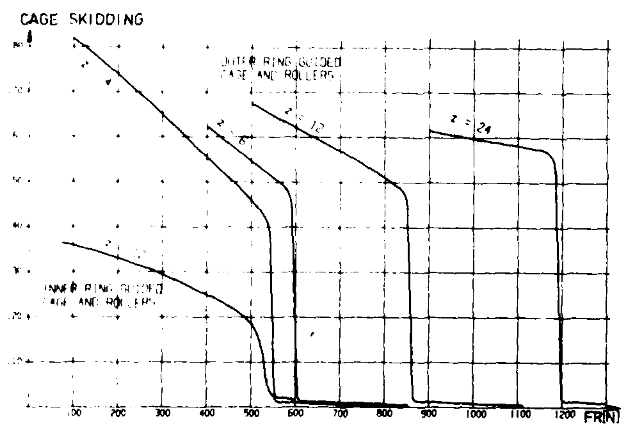


Fig 5 CAGE SKIDDING VS RADIAL LOAD

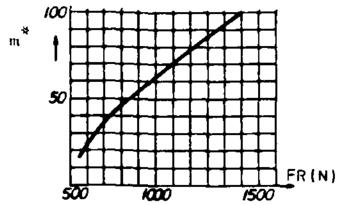


Fig. 6 CRITICAL LOAD VS RELATIVE ROLLER MASS

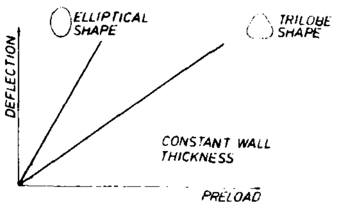


Fig. 7 STIFFNESS OF OUT OF ROUND RINGS

If the critical radial load cannot be assured at all times by external loads the out of round shape of the outer ring raceway may substitute for it. The trilobe shape, while stiffer than the elliptical for the same amount of deflection (Fig. 7), has the added advantage of centering the rotor.

INTERACTION OF ROLLER END FACE AND GUIDE FLANGE

In order to minimize friction, generation of heat and power loss the interaction betweeen the roller end face and the guide flange deserves close attention. The contact is essentially sliding and takes place at speeds in the order of magnitude of the cage speed. The contacting surface should therefore be smooth and uniform. To reduce friction the contact geometry should be conducive to the formation of a hydrodynamic lubricating film. In addition the forces needed to guide the rollers should be as small as possible .

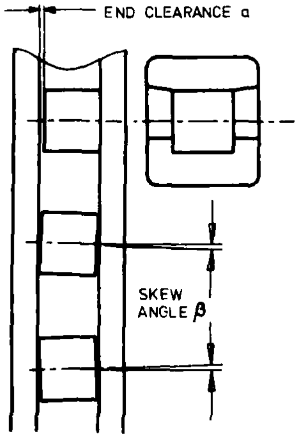
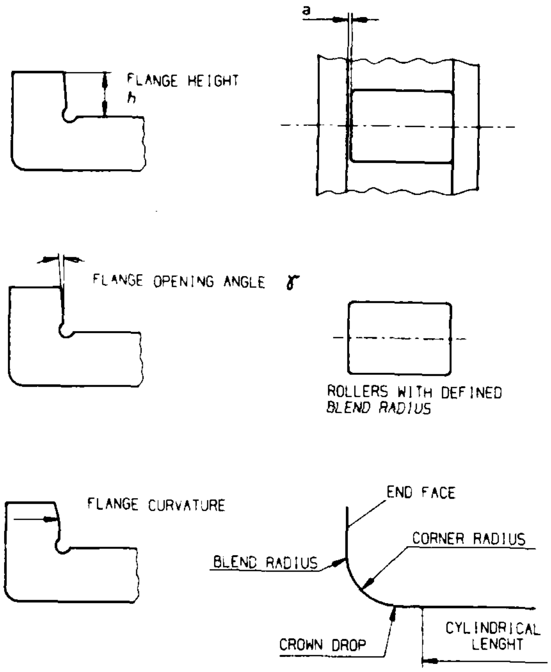


Fig 8 ROLLER GUIDANCE



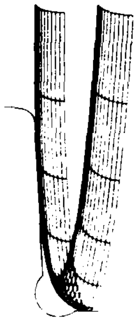


Fig 9 ROLLER END TO GUIDE FLANGE CONTACT, SKEWED POSITION

Fig 10 GUIDE FLANGE, ROLLER END CONTOUR

Fig. 8 shows the rollers within their raceway channel formed by the inner or outer raceway itself and its adjacent guide flanges. The axis of rotation of the roller shown in the top position is aligned in parallel to the axis of the bearing. The end clearance "a" is necessary to accommodate unavoidable tolerances such as the axial runout of the guide flanges and the roller end faces, to accomodate permanent or temporary deflections of the bearing ring in axial direction and to leave room for the formation of a lubricating film. In actual bearing operation the rollers will maintain this parallel position only if the raceway(s) and/or the rollers are (slightly) tapered and only as long as the roller is in contact with both races, i.e. only within the loaded zone of the bearing. At all other times the rollers will run in more or less skewed position between the two maximum skew angles $\pm\beta$. The maximum skew angle β is being determined by the points of contact between the roller end faces and guide flanges. The actual skew angle of the roller fluctuates rapidly depending on the momentary balance on the moments mentioned below. The resultant wobble of the roller modulates the orbital motion of the roller around the centerline of the outer ring raceway.

When running in a skewed position rollers with flat end faces willallways contact the guide flange with that area of the end face where it meets the corner radius of the roller (Fig. 9). It is therefore most important to blend by proper manufacturing techniques the otherwise sharp edge formed by the intersection of the flat roller end face and the corner radius of the roller (Fig. 10). To reduce friction and avoid wear this area of the roller end face may contact the guide flange only within the guide flange surface always avoiding the risk of contacting the edge formed by the guide flange surface and the adjacent cylindrical portion of the ring. The diagramm in Fig. 10 shows the minimum flange height h needed to contain the point of contact within the flange surface. For guide flanges which are perpendicular to the raceway surface, that is with a flange opening angle $\gamma = 0°$, the minimum flange height is very high even if the roller end clearance is reduced to impractical and dangerous levels. The large flange height also indicates that the sliding velocity at the point of contact is very high. With flange opening angles γ between 10 and 40 minutes the flange height is being reduced to more practical values. This can be further improved by using a crowned flange surface as shown in Fig.10. With this the flange height can be reduced to dimensions which not only limit the sliding speed but also facilitate the design of bearings, which can be disassembled by dropping the rollers away from the raceway until they clear the flange height and by removing the complement of rollers and the cage in axial direction. This design feature is desirable in the interest of inspectability of the raceway and flange surfaces. The resultant skew angle β is given in Fig. 11. This diagramm assumes that the condition of minimum flange height is satisfied.

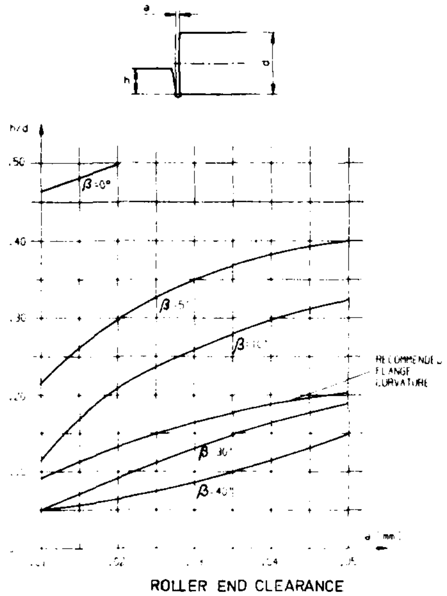


Fig 11 SKEW ANGLE β

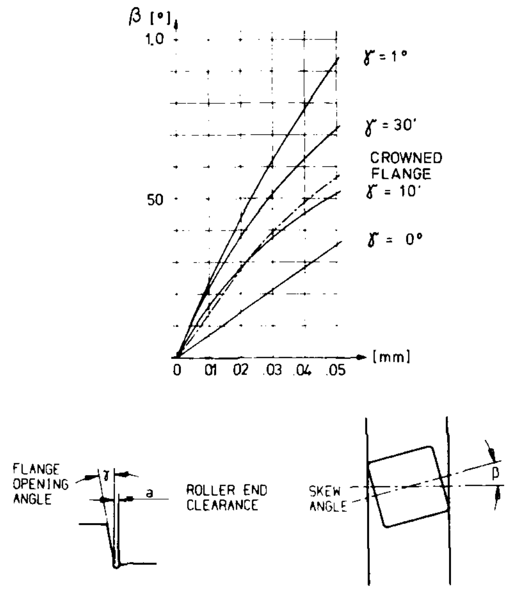


Fig.12 SKEW ANGLE β ROLLER END CLEARANCE a

Using a convex crowned roller end face the resultant skew angles β would become even larger, thus assuming values which become intolerable for very high speed bearings from the point of view of load and speed distribution between rollers and outer ring raceway and of the resultant heat generation.

The lubricant may also contribute to the driving or breaking action on cage and rollers by exchanging its impuls, i.e. by the particles of the lubricant being either retarded or accelerated when they first come in contact with cage and rollers. The amount of lubricant introduced into the bearing through under-race-lubrication (E,D and F in Fig.2) always has some driving effect while the lubricant introduced via external jets has some breaking effect. The exchange of impuls at first contact is to be distinguished from the friction losses caused by viscous drag and random churning.

In summary it may be said that to reduce friction the cage and rollers should be guided by the inner ring, the flange surface should either be tapered with a small opening angle γ or preferably crowned, while the roller end face should be flat with great attention being given to the corner blend and that under-race-lubrication should be used.

LIMITING THE ROLLER GUIDING FORCES

As already described above the cylindrical rollers have in general a tendency to depart from the position of parallel alignment of their centerline with that of the bearing. They tend to skew with the skew angle being limited by the end faces contacting the guide flanges. In order to reduce friction and avoid wear at the point of contact the contact

forces must be reduced. To analyse these forces qualitatively the moments around an axis perpendicular to the centerline and to the inner and outer ring raceways should be considered. They fall into the following two categories:

a) Acting moments (those tending to increase the contact force)

1. Moment caused by a dynamic roller imbalance (Fig. 13)

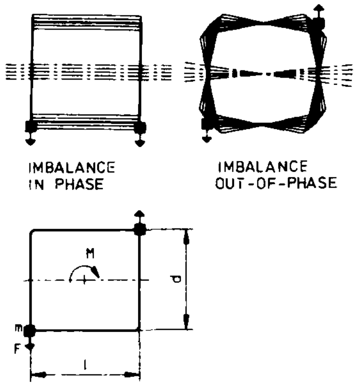


Fig 13 ROLLER IMBALANCE

Roller imbalance exists if the axis of inertia of the roller deviates from the centerline of the roller. If the axis of inertia is parallel to the centerline, however, it does not produce a moment (imbalance in phase). If the axis of inertia is at an angle with the centerline, a moment is produced around an axis that is perpendicular to the centerline. Assuming that two imbalance masses m are positioned at opposite ends of the roller and at 180° from each other (as in Fig. 13) and with a stationary outer ring and assumed epicyclic speed of the roller the moment is

$$M = F x l = 1{,}37 \cdot 10^{-9}\, m \cdot d \cdot N^2 \left[\frac{d^2}{T^2} - 2 + \frac{T^2}{d^2}\right] \cdot l$$

2. Gyroscopic Moment

With roller dimensions $d/l \leq 1.15$ the centerline of the roller is not a stable axis of rotation. Any deviation of the axis of rotation from the centerline such as a small skew angle caused by roller imbalance will produce a gyroscopic moment which tends to shift the axis of rotation to one perpendicular to the centerline.

$$M = 1{,}71 \cdot 10^{-10}\, m \cdot N^2 \left[\frac{T^2}{d^2} - 2 + \frac{d^2}{T^2}\right] \cdot \left[\frac{2}{3} l^2 - \frac{d^2}{2}\right] \sin 2\beta$$

b) Reaction Moments

1. Friction Moment

In the unloaded zone of the bearing with the roller orbital speed being assumed to be the epicyclic cage speed the friction moment produced in the contact area between roller and outer ring raceway and counteracting the tendency of the roller to skew is

$M = F \times \mu \times l$

where F is the centrifugal force acting on the roller, μ = .01 being the coefficiant of friction at full EHD-film lubrication.

For rollers with $d/l > 1.15$ the gyroscopic moment (above) becomes a reaction moment.

2. Moment Due To Cage Contact

In the loaded zone of the bearing the rollers are driving the cage by impinging against the forwardly locatead cross-web. When they enter the unloaded zone they either continue to drive or they start retarding the cage after a short period of coasting depending on whether they are inner ring or outer ring guided. With the web to web distance within the cage pocket always allowing for a larger skew angle than that determined by the guide flanges, the rollers will always be in contact only with one of the cross-webs at any given time. The contact forces exert a counter rotating moment on the rollers regardless of whether they are lagging or leading. The dog bone shaped pattern seen on the cross webs is vividly illustrating the roller to cage contact in positions determined by the range of skew angles limited by β.

To reduce friction and wear between rollers, guide flanges and cage, as well as an unfavourable stress distribution in the outer ring raceway the imbalance of rollers must be limited to very small values and the maximum skew angle should be reduced as much as possible. This can be done by defining the guide flanges as described in the previous chapter and also by reducing the end clearance to the smallest possible but safe value.

Fig 14

With the standard roller configuration with length equal its diameter (square rollers) everything possible should be done, to reduce the imbalance of the rollers. The rollers of high speed bearings may be inspected for imbalance by using the FAG-Schenck MGR 25 balancing machine shown in Fig.14 [2]. With ground roller corners and proper manufacturing and inspection techniques the value of the imbalance mass may be kept below $m < .01$ g.

With this and small skew angles the excentric roller end wear may be avoided as long as a lubricating film can be maintained at the point of contact between rollers and guide flange. If excentric roller end wear develops it adds to the already existing imbalance which leads to increasing wear rate and ultimate bearing failure.

REDUCED CENTRIFUGAL LOADING BY USE OF CERAMIC BALLS AND ROLLERS

The load carrying capacity of high speed bearings is partly being absorbed by the centrifugal load exerted on the outer ring raceway by the rolling elements which are orbiting at cage speed around the center of the bearing. This internal load increases in proportion with N^2. The Hertzian stress p_0 being proportional to the 3rd root of the raceway loading ($\sqrt[3]{F}$) for point contact, the portion of the contact pressure caused by centrifugal loading increases with $N^{2/3}$. At speeds leading to a radial acceleration TxN^2 > 1000 g the internal loading due to centrifugal load becomes a susceptible portion of the bearing load. As seen in Fig.1 if the pitch diameter is T=120mm this may be the case for a relative small TxN = $2.0x10^6$ mm/min.

The increase in contact pressure in the outer ring raceway caused by centrifugal loading reduces the fatigue life of a life limited bearing or may boost the contact pressure beyond the practical endurance limit mentioned in [3], changing the respective bearing from one with unlimited fatigue life expectancy to a fatigue limited bearing. In mainshaft positions the negative effect of centrifugal loading is more detrimental with ball bearings than with roller bearings because of the higher external loading and stress concentration in ball bearings.

There have been numerous attemps to reduce the negative influence of centrifugal loading by reducing the mass or the density of the rolling elements. In the late sixties this was done primarily by developing hollow rolling elements made of the customary aero engine bearing steels [4,5]. In order to achieve the desired reduction in centrifugal loading the wall thickness had to be reduced to the point where bending fatigue starting from the bore of the hollow rolling elements limited the life to values comparable to those of solid rolling elements or even below.

In the mid-seventies the silicon nitride emerged as the ceramic material combining low density ρ = 3.20 g/cm^3 with excellent material properties such as hardness, cleanliness, fine and uniform structure, good surface texture and so on. Major technical drawbacks are the high modulus of elasticity (Young's modulus) of E = 315 GPa and a smaller Poisson ratio of ν = .26 [6].

To minimize the disadvantage and maximize the advantage of using silicon nitride in high speed bearings only the rolling elements should be made of the ceramic material while the rings should continue to be made of steel such as the well proven AISI M50 VIM VAR. This not only reduces the stress concentration compared to a fully ceramic bearing but also avoids the problems associated with connecting the fully ceramic rings with the surrounding parts of the engine. Operating experience also shows that hybrid bearings can better absorb external or internally generated debris and have a lower rate of damage propagation than fully ceramic bearings.

The advantage of the lower density of silicon nitride rolling elements is partly compensated by the higher stress concentration caused by the higher Young's modulus and lower Poisson ratio. To achieve the same stress distribution as with all-steel bearings it would be necessary in ball bearings to increase the conformity of the races beyond the values customarily used with all-steel bearings. This, however, is possible only to a small degree due to the limitations of the manufacturing processes.

In order to realize and take advantage of the potential improvements of silicon nitride (Si3N4) rolling elements in high speed bearings the designer needs two things first:

- methods of predicting the stress distribution and rolling fatigue life (Calculation Methods)

and

- test results on rolling contact fatigue which relate to the existing experience with all-steel bearings (Test Results).

CALCULATION METHODS

Unfortunately it is not as yet possible to determine the dynamic load carrying capacity C of a hybrid bearing by simple means. We have used instead the following method taking advantage of the computer programs available at FAG for this end:

First the contact pressure constant c_p [7] is being determined for an angular contact ball bearing having ceramic balls by using the program FAG 035. Then an all-steel bearing with the same internal geometry is being calculated by varying (increasing) the inner and outer ring raceway curvatures until the same contact pressure constants c_p are found as with the hybrid bearing. The dynamic load carrying capacity of the hybrid bearing can now be calculated by means of program FAG 032 using the larger curvatures determined in the previous step. This method presumes that the same failure mode exist for both materials.

Tne dynamic equivalent load and the maximum contact pressure may now be calculated by means of program T 291 (FAG's high speed rolling fatigue program using standardized constants and exponents).

For a given bearing load the contact pressure at the outer ring raceway (Fig. 15) and the fatigue life in number of revolutions (Fig. 16) can now be plotted against the inner ring speed. In the example shown in the diagrams the external radial load was assumed to be constant at Fr = 300 N. The external axial load Fa was selected at 500, 1000 and 2000 N. By comparing the curves for the all-steel bearing with those for the hybrid bearing one can see that the fatigue life of the all-steel bearing is higher at small speeds. At a certain threshold speed the fatigue life curve of the hybrid bearing intersects that of the all-steel bearing leading to an increasing fatigue life advantage of the hybrid bearing for all speeds beyond the threshold speed. It can also be seen that the threshold speed moves towards higher values if the external loading is increased. This is because the centrifugal load is then a smaller fraction of the total loading. It can be seen that for the bearing selected and at an inner ring speed N = 60000 min^{-1} (this corresponds to TxN = 3.4 x 10^6 mm/min) the hybrid bearing will lead to fatigue life improvements even at the highest external load.

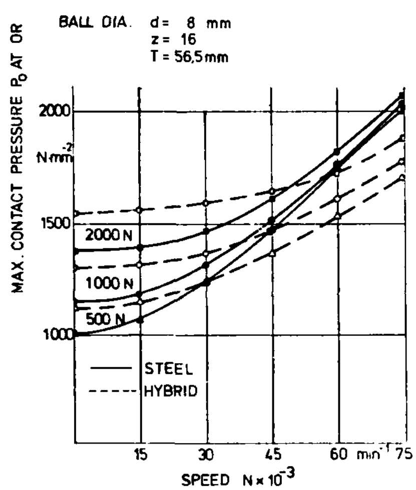


Fig.15 CONTACT PRESSURE AS A FUNCTION OF SPEED

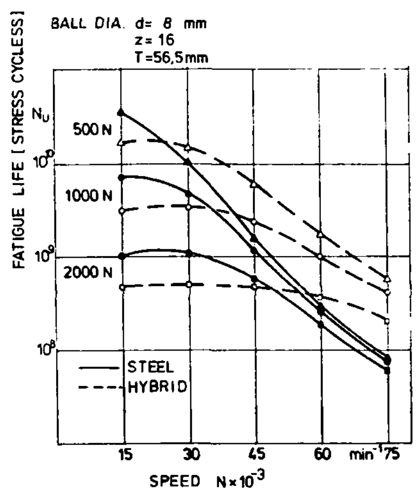


Fig.16 FATIGUE LIFE AS A FUNCTION OF SPEED

TEST RESULTS

The fatigue life testing reported 1980 in [8] had shown that at a contact pressure p = 2.9 GPa the fatigue life of hot pressed silicon nitride balls was the same as that of high grade steel balls. The fatigue life of the steel balls, however, was 30 times the calculated fatigue life (unfactored life).

Since then the testing continued with a number of production lots of hot pressed silicon nitride balls under the same conditions. The results of the various lots ranged between 15 to 50 times the calculated fatigue life (unfactored life). This lot to lot variation is consistent with the results obtained for individual lots of high grade aircraft bearings. With some advances in process control during the manufacture of the powder, its hot pressing, the manufacture and finishing of balls as well as with improved inspection methods, the results of recent lots have a tendency to range rowards the higher values.

It seems that the advances in non-destructive testing of the balls have led to the elimination of candidates for early failure, thus increasing the slope of the Weibull distribution and boosting the 10 per cent probability life.

The inspection methods are the following

a) Destructive

- SEM inspection of the fracture surface
- micrographs at 1000x and rating of metallic and non-metallic inclusions by size, number and distribution.

b) Non-Destructive

- ultrasonic inspection of the blanks
- microradiographic inspection of the blanks
- special fluourescent dye penetrant inspection of the finished balls
- ultrasonic inspection of the finished balls by the FAG SEROSONIC (registered trade mark) method [9].

Because of the high manufacturing and inspection standards and the resultant lot to lot repeatability the use of hot pressed silicon nitride rolling elements, particularly balls, may already be considered as a practical solution for high speed bearing applications. Because of the high cost of these rolling elements they are at present limited to military applications such as that reported in [10]. Based on the cost reductions due to learning curve, quantity increase, and possibly improved manufacturing technology (near - net - shape hot isostatic pressing) the cost of silicon nitride rolling elements will come down rapidly so that their widespread use in high speed applications in new engines, be they military or commercial, is easily predictable for the near future.

IMPROVED FRACTURE TOUGHNESS WITH EXISTING MATERIALS

Increasing inner ring speeds necessitate under-race-cooling and lubrication leading to intricate shapes. These are difficult to machine in such a way that high notch factors are always avoided. The high inner ring speeds also induce high hoop stresses due to centrifugal load which are further increased by the interference fit with the shaft and superimposed by high frequency bending stresses due to preloading of the cylindrical roller bearings or due to thrust loads transmitted by angular contact ball bearings. As a result a number of inner ring fractures have occurred.

The main avenue for solving this problem is to increase the fracture toughness of the inner ring by using existing case hardening materials such as RBD(DIN X 20 W Cr 10 3) or by developing new ones such as AISI M50 NIL. These case hardening materials are expected to considerably expand the speed range of safe operation. On the other hand they necessitate substantial development work to qualify for widespread use. Considering the necessity of expanding the speed range, the effort is no doubt worthwhile.

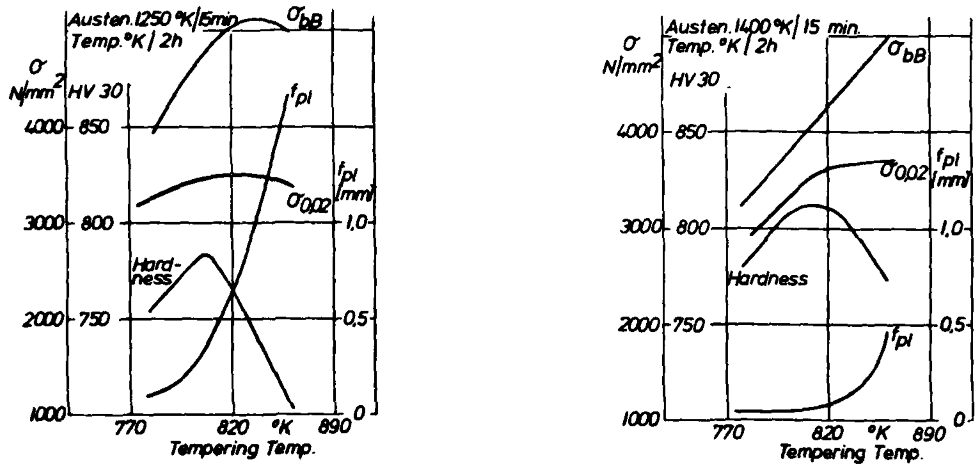


Fig. 17 INFLUENCE OF HEAT TREATMENT ON MATERIAL PROPERTIES OF M50

For high speed applications having operating conditions bordering to the danger area it may be convenient to prudently solve the problem by less costly and time consuming changes. This could be done by changing the heat treatment for otherwise conventionally through hardened AISI M50 in favor of increased ductility. As already shown in [11] and [12] the heat treatment of through hardening M50 may be optimized for greater toughness while still maintaining sufficiently high hardness and cyclic yield strength. Fig.17 shows, by way of example, the pronounced improvement in plastic deflection produced by a relatively slight reduction of the austenitizing temperature.

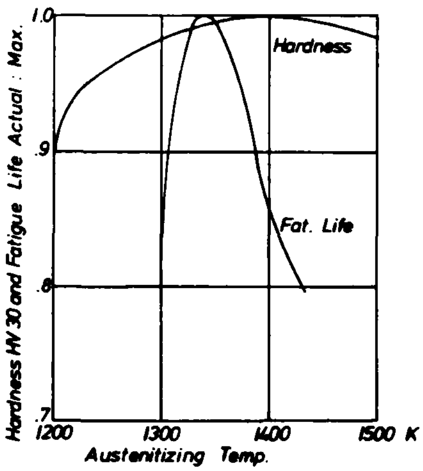


Fig.18 Relative Life and Hardness versus Austenitizing Temperature

Fig.18 shows moreover that for actual operating conditions, i. e. contact pressure p_0<2GPa and in an environment of practical but less than ideal lubrication conditions (hard particle size > film thickness) the maximum fatigue life is achieved at austenitizing temperatures below those needed for maximum hardness. With the reduced emphasis on the conventionally calculated fatigue life (unfactored life) suggested by [8] for applications with moderate contact pressures the benefit to be gained by optimizing the heat treatment in favor of improved plastic deformation outweighs the somewhat reduced wear resistance due to the lower hardness. The optimized heat treatment requires that the present hardness range of 60 to 64 HRC be altered to 57 to 59 HRC.

The optimized heat treatment cannot be expected to match the improvement to be gained by the case hardening steels but increases the margin of safety against inner ring fracture in a timely and relatively simple way.

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A STUDY OF THE POTENTIAL BENEFITS ASSOCIATED WITH THE DEVELOPMENT OF A DEDICATED HELICOPTER TRANSMISSION LUBRICANT

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SUMMARY

A common oil is now used in both the engines and transmissions of virtually all U.S. military helicopters. While this provides significant logistic advantages, these advantages are attained only by compromising the optimization of the oil for either system.

This paper summarizes the results of two studies undertaken to determine what benefits would accrue through the development of a special oil tailored specifically to meet the unique requirements of high-speed, heavily loaded helicopter transmission systems. These studies, conducted independently by two major helicopter manufacturers under the direction of the Naval Air Propulsion Center, addressed specific problem areas as related to typical production aircraft in order to reach well-documented conclusions. In addition, the effect of the availability of such a special gearbox lubricant on the development of other advanced-technology components was evaluated and documented.

BACKGROUND

Present military helicopter transmissions are required to use either MIL-L-7808 or MIL-L-23699 turbine engine oils. These lubricants contain synthetic esters formulated to the specific requirements of modern turbine engines. Extreme-pressure (EP) additives, desirable for gearing, are not used and low lubricant viscosity is a compromise to permit its use across an extremely wide temperature range.

For many years, despite the necessary tradeoffs, these lubricants have functioned very well in both applications. This has been accomplished, in the case of the gearboxes, through the use of design techniques which produced a well-functioning product within the restrictions imposed, one of which was the lubricant specification. Recently, however, the helicopter community has expressed the opinion that the performance characteristics of the MIL-L-23699 and MIL-L-7808 lubricants can no longer be tolerated for their heavily loaded power drive system if further improvements in reliability, specific weight, and size are to be obtained.

This paper is a brief, cumulative synopsis of the results of two Navy-sponsored programs (References 1 and 2) which investigated the potential benefits that might be obtained from the development of a new, dedicated transmission lubricant. The results of these studies indicate that the advantages could be quite significant.

BASELINE TRANSMISSION SELECTION

In order to evaluate the potential benefits of a new oil, an adequate frame of reference is required. Two current production helicopter main rotor transmissions were therefore selected to form the basis of these evaluations.

The production Black Hawk transmission (Figure 1) was selected as one of the baselines for comparing and examining lubricant properties and their effects on transmission performance. This gearbox represents current state-of-the-art design for helicopter transmissions and will be used extensively by both the Army and Navy. The other baseline transmission was the CH-47 Chinook forward rotor transmission (Figure 2). This aircraft has a long, successful operating history and is now being upgraded from the C to the D version. The upgrade includes redesigned main rotor gearboxes; thus this baseline is representative of both a time-tested design and an extension to the latest *state-of-the-art technology*.

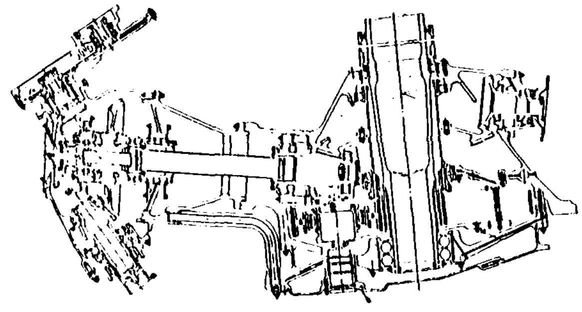

Figure 1. Cross Section Through Main Transmission of UH-60 Black Hawk Helicopter

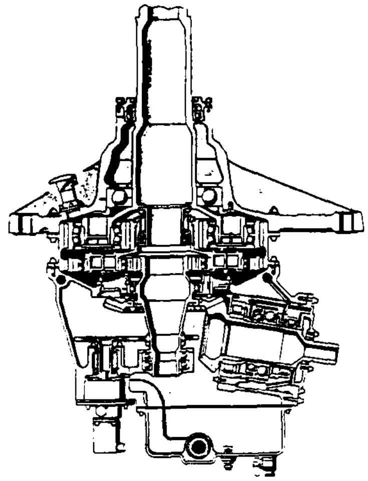

Figure 2. Cross Section Through Forward Transmission of CH-47 Chinook Helicopter

LUBRICANT PROPERTIES AND PERFORMANCE CHARACTERISTICS

Many physical and chemical lubricant properties have been defined to allow for comparisons between different lubricants and to establish requirements for producing a lubricant. Similarly, evaluation and performance tests have been established to provide a measure of the lubricants' performance when operating under the prescribed conditions. Lubricant properties and evaluation or performance tests which are typically considered for aircraft engine oils are listed in Tables I and II, respectively. For this study, our main concern is the effect of the lubricant on transmission design and performance at both present and high operating temperatures. For this reason, only those properties which are dependent on temperature or which significantly affect the design of the transmission were considered. These properties are denoted with an asterisk (*) in Tables I and II.

TABLE I. PHYSICAL AND CHEMICAL PROPERTIES

Lubricant	Thermal
Type	Specific heat*
Color	Thermal conductivity
	Expansion coefficient
Density	Flammability
Specific gravity*	Flash point*
Viscosity	Fire point
Kinematic*	Autoignition point
Absolute	Volatility
Pressure	Vapor pressure
Viscosity index (VI)	Evaporation loss
Pour point*	Distillation
	Acidity
	Neutralization number

*Lubricant parameters considered in selection of advanced lubricant

TABLE II. PERFORMANCE AND EVALUATION TESTS

Elastomer compatibility	Contamination test
Fluid compatibility	Four-ball EP test
Foaming test	Ball-bearing test
Corrosion test	Ryder gear test*
Deposition test*	Engine/transmission test
Thermal stability*	FZG gear test*
Corrosion and oxidation stability*	Demulsibility test*
Storage stability	Miscibility/compatibility
	Evaporation

*Lubricant parameters considered in selection of advanced lubricant

The properties of the lubricant define the operating temperature range and are used to predict, through analyses, the elastohydrodynamic (EHD) film thickness for each bearing contact and gear mesh, the overall system efficiency, and the scoring hazard of the gearbox. The effect of lubrication on reliability is generally considered to relate to the elastohydrodynamic (EHD) film thickness developed by the lubricant at the bearing rolling element-to-race contacts and the gear tooth contacts. Low-EHD films promote surface distress at the contact points in the form of pitting, superficial pitting, frosting, or gross wear.

Efficiency or power loss is a measure of the amount of power which is no longer available to perform useful work and the required cooling to maintain a given gearbox stabilization temperature. In the Black Hawk main transmission, for example, an efficiency change of 0.5 percent represents about 100 pounds of lost payload and approximately 75 additional pounds are required to provide external cooling.

With this background in mind it is interesting to evaluate some typical, specific lubricant-related parameters and the improvements which may be anticipated if a new oil is developed.

Viscosity

Visual examination of components, particularly bearings, confirms the role of a thicker lubricant film in extending life. These components, after equivalently long periods of time, exhibit clean, highly polished contact surfaces relatively free from damage. Conversely, components lubricated with MIL-L-23699 or MIL-L-7808 show evidence of surface damage due to low EHD film thickness and particles suspended in the oil. Each small damage point is a potential pitting initiation site; thus the increase in statistical life that is obtained with the higher viscosity oils is not hard to understand.

While all components in the gearbox will benefit from a higher viscosity oil, it is typically the bearings which show the greatest life improvement.

In order to provide some hard data on the bearing life improvement which can be expected due to a higher viscosity, we can consider the effect of a higher viscosity fluid on two bearings in the baseline gearbox shown in Figure 2. The larger roller bearing on the input spiral bevel pinion shown in Figure 2 is a typical, heavily loaded, moderate-speed helicopter transmission bearing. The first-stage planet bearing, also shown in Figure 2, is a typical, heavily loaded, low-speed helicopter bearing. Both bearings have experienced some operational difficulties, but the planet bearing has been particularly troublesome. Although its calculated B-10 life is about 1,800 hours, actual in-service experience shows a B-10 life of less than 800 hours.

Using the methods defined in Reference 3, we can evaluate the effect of the elastohydrodynamic film parameter on the life of these bearings. Our basis of comparison will be MIL-L-23699 oil with a viscosity of 5.25 centistokes (cs) at 210°F; we will assume that the new oil will have a viscosity of 10.00 cs at 210°F.

As Figure 3 shows, the probability of lubricant-related surface distress occurring decreases with increasing percentage of film formation. For our moderate-speed bearing, the percentage of full EHD film formation increases from 62 percent, which is barely out of the "region of possible surface distress...", to 94 percent, practically full development. On the other hand, the low-speed bearing shows a percentage of film formation with MIL-L-23699 oil of virtually zero. Changing the oil will bring this bearing to approximately the same percentage of full film as the higher speed spiral bevel bearing with MIL-L-23699 oil.

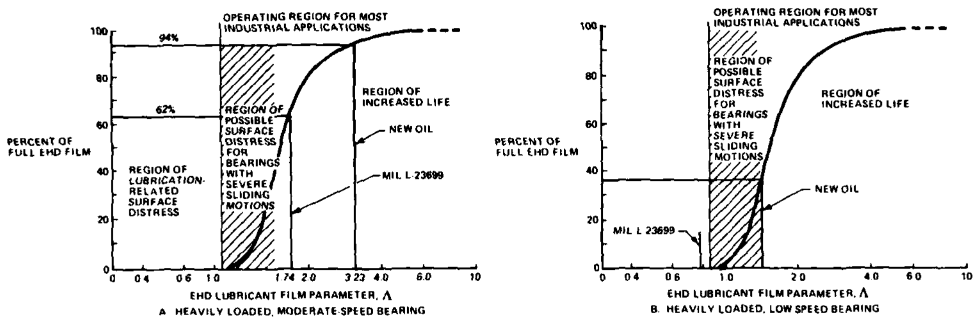

Figure 3. Percentage of Full Elastohydrodynamic Film Development as a Function of Lambda

For both bearings, an improvement in life, as shown in Table III, can be expected due to the new oil.

TABLE III. BEARING LIFE IMPROVEMENT DUE TO NEW OIL

Bearing	Oil	Lambda, Λ	Expected Life Increase (%)
Input spiral bevel pinion (heavy load, moderate speed)	MIL-L-23699	1.74	—
	New	3.23	46
First-stage planet (heavy load, low speed)	MIL-L-23699	0.60	—
	New	1.43	500

Specific Heat and Gravity

The amount of heat removed from the dynamic components of a gearbox due to the oil flow is given by:

$$q = wc\,\Delta T, \quad (1)$$

where w = oil flow, lb/hr

q = heat, Btu/hr

ΔT = temperature difference, °F

c = specific heat, Btu/lb/°F.

The amount of heat removed from the transmission can be increased by increasing any of the parameters in this equation, but only the first two, oil flow and specific heat, are unique properties of the oil. If the density (specific gravity) of the oil and its flow rate remain unchanged, an improvement in the heat removal rate can be obtained by increasing the specific heat. Decreasing the specific gravity of the oil while maintaining the same specific heat will require a higher flow rate in order to effect the same heat removal rate. For example, if the specific gravity were cut in half, the flow rate would have to double to maintain a constant heat removal rate. Since sump size is dictated by flow rate, the sump capacity would also double. This would actually result in a weight increase since, although the total weight of the oil in the sump will not increase, its larger volume would require an increase in sump size. This being the case, there is no advantage to be obtained in decreasing the specific gravity (density) of the oil.

In reviewing equation 1, it should be obvious that the parameter of greatest interest is the product of oil flow and specific heat. Considering either specific gravity (i.e., density) or specific heat alone could lead to some false conclusions, since it is the product of these parameters that is important. For this reason our ensuing discussion will consider both together. In those instances where one is discussed separately, the implication is that the other is held constant. If this product could be improved by 50 percent, then theoretically, at least, the oil flow and sump capacities required to maintain the same ΔT would also be reduced. For example, such a change for the Chinook would result in a weight saving of about 22 pounds for the baseline gearbox alone.

At first glance an increase of 50 percent may seem ambitious. Consideration of some data presented in Reference 4, however, will show that this is not necessarily so. As part of an efficiency study on an OH-58 main rotor gearbox, Mitchell and Coy measured the specific heat of several candidate lubricating oils, including four brands of MIL-L-23699 and one of MIL-L-7808. Their results, summarized in Table IV, show a drastic difference in specific heat among these oils.

TABLE IV. MEASURED PROPERTIES OF HELICOPTER TRANSMISSION OILS AT 212°F

Oil	Specification	Specific Heat, C_p* (Btu/lb/°F)	Density, ρ** (lb/gal)	$C_p \times \rho$
A	MIL-L-23699	0.32	8.21	2.63
B	MIL-L-23699	0.34	8.13	2.76
C	MIL-L-23699	0.47	7.96	3.74
D	MIL-L-23699	0.49	8.01	3.92
E	MIL-L-7808	0.30	7.27	2.33

*Per ANSI/ASTM Specification D-3947-80
**Per ANSI/ASTM Specification D-1481

Load Rating

Although it is convenient to think of the load rating of a transmission as a single number, such a simplistic view is far from the case. At the very least, the load rating must be predicated on several other factors such as environment, lubricant, internal temperature, etc. The structural integrity of the transmission, like any other machine element, is the first concern. Toward this end, gear tooth bending stress, shaft and housing stresses, and bearing capacities must be evaluated. Beyond this requirement, however, the durability of the unit must be evaluated. The structural integrity is essentially unaffected by the lubricant used, but both durability and scoring hazard are very much a function of the lubricant employed.

We have already noted the effect of lubricant viscosity on the apparent B-10 life of rolling-element bearings. Although, strictly speaking, the increased viscosity can provide a load capacity improvement, it acts more as a life capacity improvement since, for a given viscosity, the film thickness is not greatly affected by reasonable changes in load.

The main effect of using an oil with improved load capacity will be on the scoring and durability ratings of the gears.

Highly loaded gears (especially bevels) sometimes score or scuff during initial break-in. Where the scuffing is severe, it is likely that the surface finish and hardness will be adversely affected. Where MIL-L-7808 or MIL-L-23699 lubricants are used with highly stressed bevel gears mounted in magnesium housings, a 5- to 10-percent scuffing rejection rate is not uncommon. As a result, transmission manufacturers are sometimes forced to use break-in lubricants to obtain consistent quality and prevent this type of surface distress. In addition, since scoring-type failures (Figure 4) occur in a very short time, they will frequently limit the overload capacity of a system. In a

fatigue environment, a short-duration overload, if properly planned for during the design stage, will cause no life reduction since fatigue is evaluated by a cumulative-damage theory. Scoring, if it is to occur at all, will occur within minutes of operation at any particular condition. This being the case, a system is generally designed for the maximum condition as if it were a continuous condition. When this margin is built into the hardware, the weight penalty is high.

CLASS SCORING | TYPE MODERATE

CAUSE:

MODERATE SCORING RESULTS WHEN LIGHT SCORING IS ALLOWED TO PROGRESS.

REMEDY:

SAME AS LIGHT SCORING. IF BASIC CAPACITY IS EXCEEDED, REDESIGN IS THE ONLY SOLUTION.

COMMENT:

IT IS OCCASIONALLY POSSIBLE TO RETURN A GEAR WITH MODERATE SCORING TO SERVICE BY HAND-POLISHING THE TEETH WITH 400- OR 500-GRIT PAPER AND OIL.

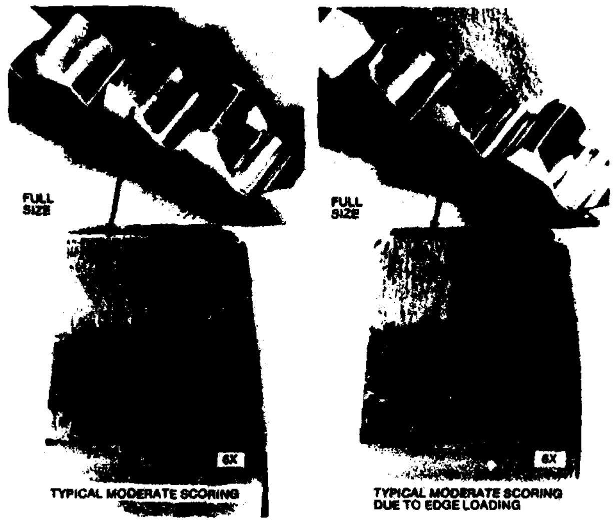

Figure 4. Typical Gear Tooth Scoring

High-hot-hardness materials, such as Vasco-X2M, have improved the balance between fatigue (bending and pitting) strength and scoring-hazard limits, but operation at higher bulk oil temperatures and/or higher loads will reduce this margin substantially. It should be obvious, then, that a large design advantage could be obtained if the design of a geared system were limited by long-term metal fatigue considerations and not by lubricant-related phenomena. In order for this to occur, the load limit defined by the lubricant capacity must be greatly improved.

Corrosion

At this time, corrosion is the leading single cause for rejection of transmission parts at overhaul for the U.S. Navy CH-46 helicopter. The latest cumulative overhaul records indicate that more than 40 percent of the gears and bearings examined at overhaul were discarded or reworked due to corrosion alone. Both the Black Hawk and the CH-46 helicopter use MIL-L-23699 oil, while the Chinook uses both MIL-L-7808 and MIL-L-23699. The incidence of corrosion on the CH-47C is considerably lower than that on the CH-46, but it still represents a considerable dollar expenditure. Since the Black Hawk is a relatively new aircraft, no historic data are available at this time.

In general, the primary problem that results from corrosion is replacement at overhaul. In order to provide some insight into the magnitude of the cost savings which may be obtained through the use of an oil with improved corrosion protection, we have reviewed the overhaul records for the CH-46 and CH-47 military fleets, as well as data from commercial operations which use the same aircraft but with other lubricants. This experience, summarized in Table V, indicates that the occurrence of corrosion in gearboxes lubricated with MIL-L-23699 oil is far greater than with either of the two other oils used. The projected cost of corrosion per flight hour, with the CH-46 fleet as a basis for comparison, was calculated using the rejection rates shown in Table V for each lubricant.

TABLE V. CORROSION EXPERIENCE FOR THREE LUBRICANTS

	Percent of Parts Corroded		
Lubricant	MIL-L-23699	MIL-L-7808	MIL-L-6082
Gears	10	2	<1
Bearings	31	7	<1

Bearing costs were calculated by averaging the cost of bearings and multiplying by the number of bearings in each transmission. It was further assumed that only half of the corroded parts would require replacement, while the remainder would be reworked and reinstalled. This is a conservative cost approach since it is seldom possible to rework a bearing (bearings account for 75 percent of main parts rejected due to corrosion), while it is often possible to rework a corroded gear. The cost of replacement only was considered; no rework costs were included in the analysis. The resultant projected costs are summarized in Table VI. The cost impact of using a lubricant with improved corrosion resistance is quite apparent.

TABLE VI. PROJECTED COST OF REPLACING CH-46 GEARS AND BEARINGS DUE TO CORROSION FOR THREE LUBRICANTS

Lubricant / Gearbox	1982 Dollars per Flight Hour		
	MIL-L-23699	MIL-L-7808	MIL-L-6082
Forward	1.82	0.29	0.16
Aft	4.45	0.54	0.29
Mix	3.94	0.62	0.30

In terms of the CH-46 fleet alone, a total cost savings of over 20 million dollars would accrue during the projected remaining life of these aircraft (about 17 years) if an oil with the corrosion-resistant properties of MIL-L-6082 were used.

While References 1 and 2 detail many more parametric evaluations which clearly define both the need for an improved lubricant and the advantages which will accrue due to its development, these few examples serve to illustrate the point.

LUBRICANT DEFINITION

In order to evaluate the potential impact of an advanced oil or oils, it is necessary to hypothetically formulate several candidate oils and to define the probability of actually achieving these formulations in practice. Two general types of oils were investigated:

1. X-Type: These fluids will permit higher reliability when operating at approximately the current temperature limits.

2. Y-Type: Higher transmission bulk oil temperatures (BOT) offer some advantage in terms of weight savings; thus this second grouping of hypothetical oils was formulated with this goal in mind.

Advanced Oils X

The first advanced oil for improved reliability was designated X1. Oil X1 was formulated to have the same physical properties and performance characteristics as MIL-L-23699 except that the viscosity at 210°F was increased from 5.0 to 10.0 centistokes and the Ryder Gear rating was increased. For this study, the Ryder Gear ratings and other performance characteristics are expressed in estimated values at BOT to allow direct comparison of ratings at various operating temperatures. As an example, the Ryder Gear rating for MIL-L-23699 is usually specified as 2,800 pounds per inch (ppi) at 165°F. With a BOT of 190°F, the Ryder Gear rating for MIL-L-23699 is estimated to be 2,600 ppi. This rating is increased to 4,000 ppi for X1 oil at 190°F. The goal with this lubricant was to at least double the gearbox MTBR and significantly reduce gear scoring while maintaining an operating temperature range from -40°F to +300°F. The oil companies consulted during this program estimated the probability of successfully formulating this oil at 70 percent due to the requirement for a -65°F pour point. This pour point is required to maintain a -40°F minimum operating temperature.

The next advanced oil, designated X2, was formulated using the same physical properties and performance characteristics as X1, except the pour point was reduced to -50°F. This change improved the probability of formulation success to 90 percent; however, the minimum-operating-temperature limit rose to -25°F.

A third oil, designated X3, was designed to further improve reliability and gear performance by increasing the viscosity at 210°F to 17.0 centistokes and increasing the Ryder Gear rating to its maximum measurable value, which is estimated to be between 4,500 and 5,500 ppi. Again, the pour point had to be decreased to -40°F in order to keep the probability of success at 90 percent. This pour point results in a minimum operating temperature limit of -10°F.

The fourth oil, designated X4, was formulated to operate at the maximum BOT of MIL-L-23699 (350°F) but with the viscosity and Ryder Gear rating of MIL-L-23699 at the present transmission BOT of 190°F. At this temperature, the ratings of the performance test are substantially reduced and the probability of successfully formulating the lubricant is only 40 percent. The chance of success is low because of uncertainty in obtaining a Ryder Gear rating of 2,600 ppi at a BOT of 350°F and of meeting the -10°F minimum-operating-temperature requirement. However, if this oil is designated as an improved-reliability lubricant and run at the present operating

conditions (BOT of 190°F), the probability of success increases to 60 percent and substantial reliability improvements can be achieved.

Advanced Oils Y

All of these lubricants, as summarized in Table VII, were designed to operate up to 600°F. Significant efforts have been made in the past to develop lubricants to operate at this temperature and, while successful formulations have been made, their performance when operated with bearings and gears has been marginal. The sole purpose in operating a helicopter transmission at these high temperatures is the significant weight saving and reduced vulnerability achieved by removal of the oil cooling system. This potential advantage will have to be carefully weighed against the probability of developing a successful oil and the probability that this oil will provide an acceptable level of reliability for the gearbox.

TABLE VII. PHYSICAL PROPERTIES AND PERFORMANCE CHARACTERISTICS OF MIL-L-23699 AND ADVANCED OILS

A. Physical Properties

Lubricant	Pour Point at 100K CS (°F)	Flash Point (°F)	Temperature (°F) Min at 13K CS	Temperature (°F) BOT	Temperature (°F) Max	Viscosity (cs) at 210°F	Viscosity (cs) at BOT	Viscosity (cs) at Max °F	Specific Gravity at BOT	Specific Heat at BOT (Btu/lb-°F)
MIL-L-23699 (production design)	-65	450	-43	190	300	5.0	6.2	2.4	0.925	0.504
Advanced Oil X										
X_1	-65	450	-40	190	300	10.0	12.5	4.5	0.60-0.95	0.40-0.50
X_2	-50	450	-25	190	300	10.0	12.8	4.2	0.80-0.95	0.40-0.50
X_3	-40	450	-12	190	300	17.0	22.5	6.6	0.80-0.95	0.40-0.50
X_4	-40	450	-10	190	300	24.0	32.0	9.2	0.80-0.95	0.40-0.50
MIL-L-23699	-65	450	-43	350	450	5.0	1.6	1.2	0.860	0.602
Advanced Oil X_4	-40	450	-10	350	450	24.0	6.2	3.4	0.72-0.90	0.45-0.55
Advanced Oil Y										
Y_1	-40	525	-3	460	600	51.0	6.2	3.4	0.70-1.0	0.48-0.58
Y_2	-40	525	-10	460	600	22.0	3.0	1.8	0.70-1.0	0.48-0.58
Y_3	-40	525	-13	460	600	11.3	1.8	1.1	0.70-1.0	0.48-0.58
Y_4	-40	525	-13	460	600	11.3	1.8	1.1	0.70-1.0	0.48-0.58

B. Performance Characteristics at BOT

Lubricant	Ryder Gear Rating (ppi)	Corrosion Oxidation Stability Test 1-10	Deposition Test 1-10	Thermal Stability 1-10	Probability of Success (%)
MIL-L-23699 (production design)	2,600	10	10	10	100
Advanced Oil X					
X_1	4,000	10	10	10	70
X_2	4,000	10	10	10	90
X_3	4,500	10	10	10	90
X_4	4,500	10	10	10	60
MIL-L-23699	1,800	8	8	8	100
Advanced Oil X_4	2,600	8	8	8	40
Advanced Oil Y					
Y_1	2,600	10	10	10	1
Y_2	1,800	7	7	7	30
Y_3	1,100	6	6	6	40
Y_4	1,100	5	5	5	50

The first oil, designated Y1, was formulated to provide the same viscosity, Ryder Gear rating, and performance characteristics at a BOT of 460°F as MIL-L-23699 has at a BOT of 190°F. The most significant problems associated with operating at these temperatures are thermal stability and oxidation. For this reason the probability of successfully formulating advanced oil Y1 is less than 1 percent. For the second oil, the viscosity at BOT was reduced to 3.0 centistokes, the Ryder Gear rating was reduced to 1,800 ppi, and the remaining performance characteristics were reduced from 10 to 7 based on a scale of 1 to 10. These changes improved the probability of formulating the oil to 30 percent. The viscosity of the third oil, Y_3, was reduced to equal the viscosity of MIL-L-23699 at 350°F. The performance characteristics were also reduced, which improved the probability of formulating the oil to 40 percent. Only by reducing the Ryder Gear rating to 1,100 ppi and the other performance characteristics to 5 could the measure of potential success in formulating the lubricant be increased to 50 percent. This oil was designated Y_4.

A summary of the physical properties and the performance characteristics for MIL-L-23699 and advanced oils X and Y is presented in Table VII, while Figure 5 shows the relationship between reliability or weight saving, and the maximum surface temperature of the Black Hawk main gearbox.

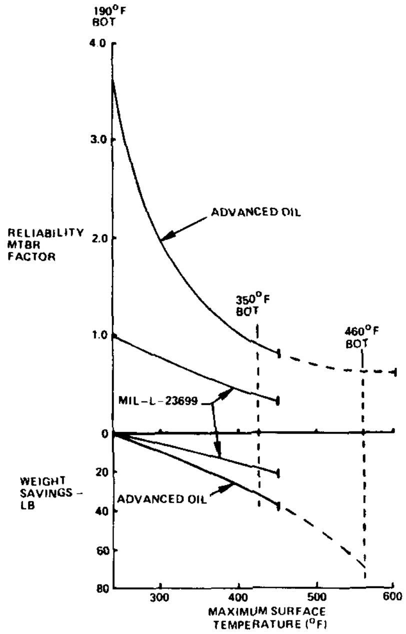

Figure 5. Effect of Advanced Lubricant on Reliability and Weight

POTENTIAL BENEFITS

The benefits associated with the development of a new oil are most clearly demonstrated by considering the specific effects of various property improvements on the operation of the two baseline transmissions. Tables VIII, IX, and X summarize these data for the Chinook and Black Hawk baselines, respectively.

CONCLUSIONS

The need for and potential benefits of a new oil specifically tailored for use in the next generation of advanced helicopter transmission systems have been clearly demonstrated and defined. Specific examples have been cited to support the major areas of concern.

RECOMMENDATION

The development of the new oil should proceed immediately. This development should be conducted with the guidance and full participation of the helicopter transmission design community.

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TABLE VIII. LUBRICANT DEFINITION AND ITS EFFECT ON THE CHINOOK BASELINE

Property	Current Level	Required Level	Benefits Anticipated
Viscosity at 212°F	5 cs	10 cs	Improved bearing life
Example:	46% increase: CH-47C forward transmission input pinion bearing (moderate speed, high load) 500% increase: CH-47C planet bearing (low speed, high load)		
Specific Heat at 212°F	Not specified	0.50 Btu/lb/°F per ASTM D3947-80	Reduced oil system weight and more consistent performance
Specific Gravity at 212°F	Not specified	0.985 per ASTM D1481	Reduced oil system weight and more consistent performance
Example:	22-lb savings on CH-47C forward transmission 22-lb savings on CH-47C aft transmission 9-lb savings on CH-47C mix box 53-lb savings, total		At specific gravity of 0.985
Load Capacity at 212°F	2,800 ppi Ryder (approx)	3,700 ppi Ryder (minimum)	(1) increased power, (2) eliminate fretting, (3) reduced weight, (4) increased life
Example:	(1) CH-47C planetary system 15% increase in bending load capacity through use of high-contact-ratio gearing 28% increase in surface load capacity (2) CH-47C sun/bevel gear integration eliminates bolted joint but increases flash temperature. (3) 19-lb (25%) reduction in weight of each CH-47C main transmission input pinion cartridge through the use of high-speed tapers (total weight savings per aircraft = 38 lb). U.S. tapers require higher-load-capacity oil. (4) 540% increase in life of bearings on CH-47C main transmission input pinion bearings possible if replaced with U.S. taper system.		
Operating Temperature	350°F	400°F	(1) reduced seal leakage, (2) increased spline misalignment capability
Example:	(1) High localized temperatures at shaft/lip seal interface result in oil breakdown and deposits forming on shaft, causing leakage. (2) Higher temperature limit will permit use of advanced lip seals at larger diameters (increased rubbing speed) and allow shaft spline L/D ratio to be reduced greatly to increase misalignment capability.		
Corrosion Resistance	Not specified	ASTM D665	Decreased operating costs
Demulsibility	Not specified	ASTM D1401	Decreased operating costs
Example:	600% reduction in cost (per flight hour) of replacing corroded gears and bearings can be achieved if corrosion resistance of MIL-L-23699 in presence of water can be brought up to that of straight mineral oil.		

TABLE IX. LUBRICANT DEFINITION AND ITS EFFECT ON THE BLACK HAWK BASELINE DESIGN PARAMETERS

A. Temperature and Elastohydrodynamic Film

Lubricant		Temperature (°F)				EHD — Λ Minimum	
		Bulk Oil	Maximum Surface	Operating Range Minimum	Operating Range Maximum	Bearings	Gears
MIL-L-23699 (production design)		190	240	-40	300	0.15	0.20
Advanced Oil X	X_1	190	240	-40	300	0.36	0.47
	X_2	190	240	-25	300	0.36	0.48
	X_3	190	240	-12	300	0.52	1.01
	X_4	190	240	-10	300	0.71	1.63
MIL-L-23699		350	425	-40	450	0.04	0.04
Advanced Oil X	X_4	350	425	-10	450	0.13	0.15
Advanced Oil Y	Y_1	460	560	- 3	600	0.12	0.13
	Y_2	460	560	-10	600	0.06	0.06
	Y_3	460	560	-13	600	0.04	0.04
	Y_4	460	560	-13	600	0.04	0.04

B. Scoring, Efficiency, and Cooling

Lubricant		Maximum Scoring Index (°F)	Efficiency Loss (%)		Total Power Loss (Btu/min)	Housing Cooling (Btu/min)	External Cooling Required (Btu/min)
			Bearings	Gears			
MIL-L-23699 (production design)		520	0.96	1.77	3,310	640	2,670
Advanced Oil X	X_1	520	1.35	1.53	3,460	640	2,820
	X_2	520	1.36	14.9	3,420	640	2,780
	X_3	520	1.76	1.05	3,380	640	2,740
	X_4	520	2.12	0.62	3,280	640	2,640
MIL-L-23699		680	0.75	2.62	4,040	2,060	1,980
Advanced Oil X	X_4	680	0.99	1.80	3,350	2,060	1,290
Advanced Oil Y	Y_1	790	0.98	1.84	3,390	3,540	0
	Y_2	790	0.76	2.36	3,750	3,540	210
	Y_3	790	0.65	2.62	3,920	3,540	380
	Y_4	790	0.65	2.62	3,920	3,540	380

TABLE X. LUBRICANT DEFINITION AND ITS EFFECT ON THE BLACK HAWK BASELINE PERFORMANCE PARAMETERS

A. Temperature and Reliability

Configuration		Lubricant		Temperature (°F) Bulk Oil	Temperature (°F) Maximum Surface	Bearings MTBR Factor	Gears Scoring Probability (%)
Present production		MIL-L-23699 (production design)		190	240	1.00	30.
Improved-reliability gearbox		Advanced oil X	X_1	190	240	1.65	2.
			X_2	190	240	1.66	2.
			X_3	190	240	2.69	0.2
			X_4	190	240	3.50	0.3
		MIL-L-23699		350	425	0.50	99.
	Reduced cooling	Advanced oil X	X_4	350	425	0.90	30.
High-temperature gearbox	No external cooling	Advanced oil Y	Y_1	460	560	0.87	30.
			Y_2	460	560	0.61	93.
			Y_3	460	560	0.50	99.
			Y_4	460	560	0.50	99.

B. Weight, Efficiency, Survivability, and Vulnerability

Configuration		Lubricant		Weight Change (lb)	Efficiency (%)	Survivability Factor	Vulnerability 1 to 10
Present production		MIL-L-23699 (production design)		0	97.27	1.0	5
Improved-reliability gearbox		Advanced oil X	X_1	4	97.12	0.9	5
			X_2	3	97.15	0.9	5
			X_3	0	97.19	1.0	5
			X_4	0	97.26	1.1	5
		MIL-L-23699		-19	96.63	0.6	6
	Reduced cooling	Advanced oil X	X_4	-37	97.21	0.6	6
High-temperature gearbox	No external cooling	Advanced oil Y	Y_1	-75	97.18	0.4	8
			Y_2	-70	96.88	0.5	8
			Y_3	-65	96.73	0.6	8
			Y_4	-65	96.73	0.6	8

AD-P005 059

Multifunctional Requirements for a Gearbox Secondary Power System in a Modern Fighter Aircraft and its Components and Interface Requirements

by

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Summary

The basic task of the gearbox in a Secondary Power System (SPS) of an aircraft is to provide and distribute mechanical energy from the Auxiliary Power Unit (APU) or from the Main Engine (ME) to the accessories, i.g. generators, hydraulic pumps and fuel pumps. The complexity as well as the functional facilities increase by incorporating further functional requirements such as main engine start, cross drive operation between redundant systems with the associated control devices, including oil supply to the accessories into the gearbox oil system.

The Tornado-SPS is one of the most modern systems flying today. Inspite of its high complexity, as a result of the extreme functional requirements, the system shows a high reliability although the aircraft has been inservice for a few years only. The experience gained with major components like gearbox housing, gearing, freewheel clutches, dry friction clutches and components of the oil system will be reported. Some aspects from this experience will be given for the design of gearboxes for future secondary power systems.

List of abbreviations

A/C	Aircraft
ATM	Air Turbine Motor
APU	Auxiliary Power Unit
C/D	Cross Drive
ECS	Environmental Control System
FBP	Fuel Backing Pump
GB	Gearbox
Ge	Generator
HP	Hydraulic Pump
IDG	Integrated Drive Generator
ME	Main Engine
PTO	Power Take Off

1. Introduction

With increasing requirements for secondary power as to hydraulic and electrical energy, a secondary power system (SPS) separately arranged from the main engines has demonstrated its advantage as to flexibility in arrangement, maintenance, reliability and life cycle cost.

Being responsible for engineering and production of the APU and the Gearboxes for the Tornado-SPS, KHD Luftfahrttechnik GmbH became a potent member of SPS specialists. The following review shows a long tradition of the company in the field of aeronautics:

- 1892 Establishment of a motor manufacturing company in Oberursel.
- 1913 Start of production of the aircraft piston engine "Gnome-Rhone", which powered the famous Fokker triplane.
- 1930 Integration of the Oberursel motor company into the Klöckner-Humboldt Deutz AG.
- 1939-1945 Development of twostroke piston engines with high supercharging and Schnürle loop scavanging (under the leadership of Dr.Ing. Schnürle).
- 1956 Start of development of small turbine engines.
- 1959 Start of manufacturing and servicing of gas turbine engines for the German Airforce.
- 1970 Start of development of the APU and the gearboxes for the Tornado-SPS.
- 1972 Entering into engineering collaboration on recuperated vehicular gasturbines with Garett, Volvo and Mack Truck.
- 1975 To date, studies into the requirements for secondary power systems for future fighter aircrafts.
- 1976 Start of development of small turbojet engines for recornaissance drones.
- 1980 Foundation of the KHD Luftfahrttechnik GmbH as a selfadministrated company which demonstrates the importance of aeronautic activities.

The subject of this paper is to give a survey of the experience gained with components of the Tornado gearboxes demonstrating the dominant factors for the reliability of the system. Because of the wide field some extracts can be given only. Some aspects for the design of SPS-components for future fighter aircrafts will be shown.

2. Description of the Tornado-SPS

According to the design philosophy in the 1960's, the *SPS for the Tornado (fig.1) was* designed as a mechanically interconnected duplex system as the scheme in fig.2 shows. In the aft section of the aircraft two gearboxes are located connected to the two main engines via flexible Power Take Off (PTO) shafts. One Integrated Drive Generator (IDG), one Hydraulic Pump (HP) and one Fuel Backing Pump are mounted to each gearbox. In addition the Auxiliary Power Unit (APU) is located on the starboard gearbox. Modules of the starboard gearbox only are a clutch between APU and gearbox and a cross drive clutch. The function of the SPS is controlled by an electronic control unit, whereby failures of essential functions are monitored at the maintenance panal.

The SPS is activated by starting the APU with an electrical starter motor which is powered from an onboard battery. After runup of the APU, the APU-clutch is activated automatically and the starboard gearbox runs up with controlled acceleration. This is required from the acceleration limit of the IDG. It is the pilot's decision to activate the cross drive clutch. While the SPS is driven by the APU, the hydraulic and electrical A/C-system can be checked and the energy supply is achieved for a sufficient long periode for standby.

Starting one of the main engines is initiated by filling of the related torque converter for the starting period . After runup, this main engine is then driving the SPS by engaging of a freewheel clutch located within the torque converter module.The second main engine can be started via the cross drive also, by filling the related torque converter. Finally the cross drive clutch is opened and both gearboxes are driven by their main engines. After a main engine flame out on ground or inflight this engine can be restarted by a cross start operation.

3. Requirements onto the components of the gearboxes of the Tornado-SPS, experience with these components and resulting benefits for future applications

Today a very broad experience base has been accumulated with the Tornado-SPS. With bench endurance running about 5000 hours and in active service more than 100000 operating hours have been achieved. For a lot of parts common in both gearboxes, the experience accounts twice. Valuable knowledge has been gained regarding the reliability of the components, assisted by defect analysis results. This knowledge will benefit the design of future systems.

Several studies have been done to derive the requirements for secondary power systems for future fighter aircrafts. An important feature is the power transmission for starting, engaging the main engines to a gearbox and a cross drive operation in the case of twin engine powered aircrafts. Different systems, including mechanical, electrical, hydraulic and pneumatic power transmission have been valued in relation to reliability, weight, efficiency, maintenance and cost. Detailed informations are given in ref.(1,2). From these evaluations a system with a mechanical transmission between main engine and gearbox (high efficiency) and a pneumatic transmission for the starting system and the cross drive (high reliability and flexibility) shows decisive future advantages. Therefore, for the next european fighter aircraft the system as per fig.3 is favoured.

In the following chapters the experience with some components of the Tornado-SPS gained from development and inservice will be reported, in particlar with regard to the benefit for future development.

3.1 Gearbox housing, especially in view of material, strength and corrosion protection

Because of the high weight of the accessories and hence their overhang moments severe structural requirements for stiffness and strength must be fulfilled for the housings of the gearboxes. To obtain a low weight the housings are made from magnesium alloy RZ 5 (AMS 4439A). For cost reduction and improvement of stiffness most of the required oil passages were integrated into the casting. This high level of technology in casting and coring is realized today by both Haley Industries Ltd., Canada and Honsel, Federal Republic of Germany. As an example fig.4 shows the starboard gearbox housing.

Due to the complexity of the housing structure and various loads such as inertial- and gear loads, extended tests had to be done to proof the stiffness, static and dynamic strength. Especially mechanical vibrations can be demonstrated experimentally only, because of the influence of the damping behaviour of the accessories. From this it is evident that vibration test must be done with original accessories.

Although magnesium alloys are very sensitive to corrosive attacks, a sufficient protection is feasible by special surface treatments. The gearbox housings are completely treated by the electrochemical HAE-process: An anodic oxydation process builds up a porous and hard surface layer which gives an excellent adhesive basis for lacquer. Proper attention to details, such as corner radii, however, has to be paid by design. When comparing aluminium and magnesium alloys, as regards to benefit of weight, it is therefore not necessary for future applications to give up the use of magnesium alloys, especially when higher strength at elevated temperatures is required.

3.2 Gearing

Different types of gears are the essential parts of gearboxes, but mainly spur gears are used. In past years the design methodology and the material properties have been developed to a high state of the art. As an example fig.5 shows a highly loaded component from the Tornado gearboxes which demonstrates the light weight design. There is no noticeable wear after 1500 operating hours, respectively after more than 10^9 contact cycles. This gear was treated by shot peening to improve the fatigue strength which allowed a reduction of the teeth length by 20%.

As a tribological aspect the lubrication oil according to UK specification DERD 2497 used for the Tornado aircraft shows a higher load carrying capability in comparison to MIL-L-23699C, which is common in the US. This is of special advantage for the secondary power system (for further data see chapter 3.6 and fig.15).

Gears used today normally are designed for a contact ratio of 1.4 to 1.7. For reasons of improving performance as to the load carrying capability, as to vibration and noise generation, future products will use gears with high contact ratio (HCR) in a significantly higher extend (ref.(3)). This tendency is assisted by the trend to higher operating speeds, in order to improve the ratio of transmitted power to the mass of the equipment.

Since 1975 HCR gears are under development at KHD's too. For example a noise reduction of more than 3 decibel (at laboratory conditions 10 decibels was obtained) and an increase of the load carrying capability of 10% and more has been demonstrated. But it should be mentioned that HCR gears are more sensitive to tolerances and therefore higher quality is required, this has to be considered when judging the life cycle costs.

3.3 Freewheel clutches

Within the Tornado gearboxes two different freewheel clutches are used, one in the APU clutch module and the other in the torque converter module of each gearbox (fig.6). In both cases the freewheel clutches are of the sprag type, the functional requirements, however, being different.

The freewheel in the APU clutch module engages with the beginning of APU clutch operation at zero speed and remains engaged while the APU is running. The freewheel in the torque converter module, however, is engaged when the gearbox is already driven by the main engine. As an important requirement for this clutch, engagement must be possible within the whole gearbox speed range without delay. This can be understood when looking to the cross drive operation mode: Assume both main engines are running with different speeds then both gearboxes are driven from that main engine which runs with higher speed. After opening the cross drive clutch now the nondriven gearbox is decellerated at a rate of about 10000 rpm/s until the freewheel clutch engages. A delay at engaging would results in a high shock torque. Therefore a big effort was required for design and testing the engaging behaviour of the freewheel clutches.

In the beginning of the engineering phase the design was made according to the recommondations of the suppliers of the freewheel clutches. During testing, several unexpected failures occured. Therefore a special testrig was developed by KHD for the investigation of the engaging behaviour within the whole speed range. The scheme of the test arrangement is shown in fig.7. By using an electronically controlled drive the freewheel clutch is engaged periodically at different speeds. At the end of each period the freewheel is disengaged and the inner shaft decellerates. The decelleration rate depends on the friction torque excited by the slipping sprags and the mass of inertia of the shaft. This friction torque represents an equivalent to the engaging force, which is produced by the centrifugal force (fig.8). This inturn is caused by the eccentricity of the centre of gravity of the sprags in relation to the centre of the radius of the outer sprag contour and also by the energizing spring.

An engaging force too low results in a delayed engaging, however, an engaging force too high produces an unacceptable wear at the inner race during the overrunning mode. Fig.9 shows the behaviour of two different sprag types. From our experience and measured as explained, an engaging friction torque of 0.1 to 0.2 Nm (value depends on sprag clutch size) was required for the torque converter freewheel clutch. Furthermore as a result from this investigation it was found that even minute wear at the outer radius of the sprags already shows an influence on the dynamic behaviour of the freewheel clutch.

Based on this investigation the arrangement of the torque converter freewheel clutch was redesigned in regard to sprag type, spring rate and lubrication. The result of this design has demonstrated high reliability with more than 1000 inservice operating hours.

In regard to the lubrication oil as a negative aspect the formation of oil sludge and sediments was found when using incompatible oil mixtures. From these a reduction of the friction coefficient at the outer race and of the ability of the sprags to engange occur and results in malfunctions.

At present, within a dedicated evaluation program, the behaviour for operating times beyond 1000 hours is under investigation regarding the influence of wear onto the dynamic behaviour and the strut angles. These activities are not only helpful within the existing Tornado program, but they are important for the design of future secondary power systems. Taken into account the trend towards higher operating speeds, the functional requirements for the freewheel clutches have to be raised.

3.4 Dry friction clutches

As already mentioned within the description of the Tornado-SPS, there are two clutches arranged in the starboard gearbox. From logistical reasons both clutches are identical (see fig.2). Caused by the limited acceleration rate of the IDG an acceleration control had been introducted for the clutch operation. Fig.10 shows the typical behaviour of the control system. Even when the IDG or the HP are being subjected to slam loading, the resulting severe torque steps are compensated by the control system such as to be of minor influence on the acceleration rate.

A relatively long slipping period is the result from the limited acceleration rate with the concequence of a high thermal load to the clutch. For example in fig.11 the heat generation of the APU clutch is shown dependent on the oil temperature. The strong increase of heat generation at low temperatures is caused on one side by the increased drag torques as an influence of oil viscosity and on the other side by an increased slippage time as a result of the required lower acceleration rate of the IDG, dependent on oil temperature. This has to be considered in the control system. Taking into account these high thermal loads the clutches were designed as dry friction clutches, because of their higher thermal capacity in comparison to wet friction clutches. Wear problems at the inner and outer splines have been solved by special coatings.

In service, however, a disadvantage was found: Dry friction clutches are sensitive to malfunctions of the control devices and mishandling. Therefore for future systems such high loaded friction clutches should be avoided, respectively their influence to system reliability must be minimized.

3.5 Hydraulic torque converter

For starting the main engine each gearbox contains a hydraulic torque converter, which normally runs empty and is filled with lubrication oil for main engine start only. The torque ratio is about 2.8 for zero output speed. According to the characteristic the input power increases with the third power of the input speed. In the event of one engine flame out inflight the normal power absorption of the torque converter for a restart via cross drive could not be handled by the driving system. Therefore a partial filling valve has been developed to reduce the power consumption of the torque converter at higher speeds. The gearbox speed is sensed by using the oil flow of the lubrication pump as an equivalent indicator. Fig.12 shows the partial filling behaviour at different speeds.

In practice a high reliability can be expected for a hydraulic torque converter. This was confirmed inservice: No failure occured until today. For a secondary power system according to fig.3 no torque converter is required, because the air turbine motor has a similar characteristic. Nevertheless taking into account reliability and cost the use of a hydraulic clutch within the train to the main engine should be investigated.

3.6 Oil system of the gearbox and its components

To fulfill the requirements in respect to clutch control, torque converter operation, lubrication and the integration of the oil systems of the IDG and of the APU a sophisticated oil system had to be realized. In addition, further requirements had to be taken into consideration: Attitude, especially inverted flight, and the oil supply to the cross drive clutch under the condition of a running port gearbox while the starboard gearbox is at rest. To meet all these requirements many valves and three different oil pumps were developed. Extensive analysis and tests were done in the field of hydraulic vibrations. Fig.13 shows some results from testing of the pressure relief valve for the clutch supply. Within this investigation the lenght of the oil metering slots and the diameter of the damping orifice were varied. The different valve modifications were tested in an operating gearbox and the hydraulic vibration was examined. As fig.13 shows, a clear relation was found between required valve damping and the lenght of the oil metering slots from which the amplification of the pressure control depends.

To prevent cavitation within the constant speed drive of the IDG a minimum absolute level is required for gearcase pressure, especially at high altitudes. For pressurization a vane pump was developed as shown in fig.14, which is used in both gearboxes. Despite of a speed of about 12000 rpm, wear could be prevented by coating of the sleeve.

For the Tornado-SPS a lubrication oil according to UK specification DERD 2497 (NATO code O-160) is used. Fig.15 shows characteristic data of different oil brands and a comparison with a typical oil of MIL-L-23699C specification. While at normal temperatures the differences as to viscosity are low, the oils according DERD 2497 shows a much higher viscosity at -40°C. This has an influence onto the drag torques and the cold start behaviour. The differences as to load carrying capability are already reported in chapter 3.2.

At rig endurance tests an acidification of the oil had been detected, but only in the port gearbox and not in the starboard gearbox. As fig.16 shows, there is a linear increase versus operating time. The endurance test accumulates defined test cycles of two hours each. Fig.17 shows the applied temperature profile, which is controlled during the test. As reason for the different behaviour of the oil in the gearboxes, port and starboard, the following explanation was found:

As already mentioned both gearboxes have a pressurization system and must be tightly

sealed therefore. The starboard gearbox must be vented during APU operation, however, to prevent an unacceptable pressure increase from the air/oil suction flow of the APU. Therefore the more volatile acids evaporates through the venting port, especially as the oil is heated up during this time. For proofing of this hypothesis a vented system at the port gearbox was introduced experimentally. The result showed a simular low acidification as in the starboard gearbox. This demonstrates that the acidification is higher in a closed system, even at moderate temperatures. With the exception that, with higher acidity rates, deposits appear on cadmium and silver plated parts, no clear statement can be given about further influences of high acid numbers. From oil analysis after the endurance testing as a result was found a lower load carrying capability in case of the higher acidification. In practice, however, no problem was found up today, by cause of lower oil temperatures in comparison to test rigs also results lower acidification.

4. Conclusions

The secondary power system is of high importance in regard to reliability and function of an aircraft. The key position belongs to the gearboxes. The reliability of a gearbox is determined by the reliability of its components. Today for the Tornado-SPS a broad experience exists. The experience with some important components has been reported in this paper. This experience will be of extreme advantage and benefit for the design of a secondary power system for future aircraft. KHD Luftfahrttechnik GmbH has continuously increased the level of technology in this demanding discipline.

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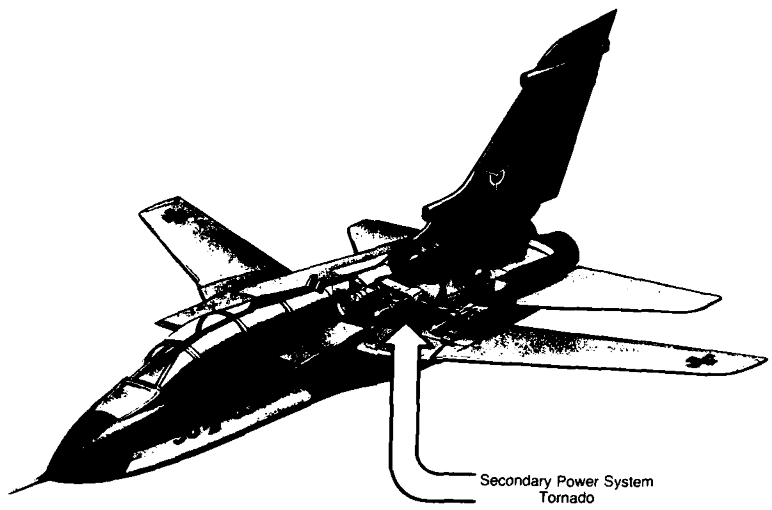

FIG. 1: ARRANGEMENT OF THE SPS WITHIN THE TORNADO AIRCRAFT

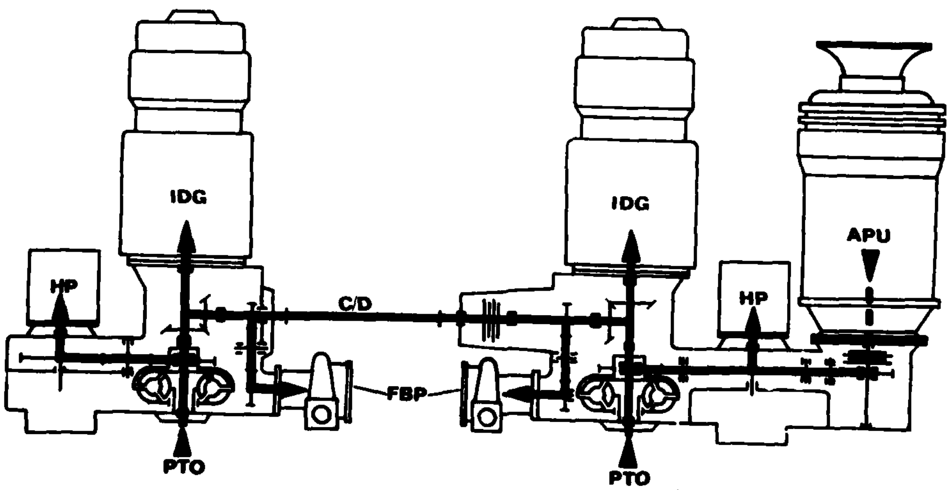

FIG. 2: SECONDARY POWER SYSTEM OF THE TORNADO AIRCRAFT

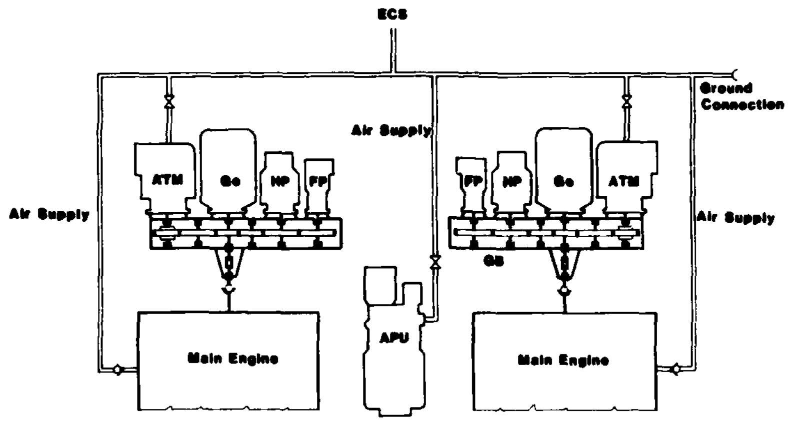

FIG. 3: SPS WITH PNEUMATIC ENERGY TRANSMISSION

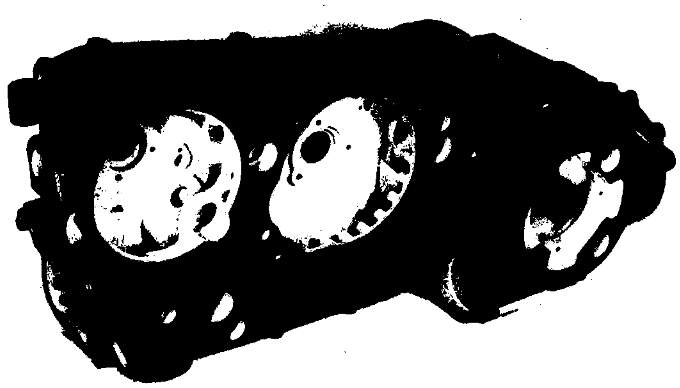

FIG. 4: HOUSING OF STARBOARD GEARBOX OF THE TORNADO SPS

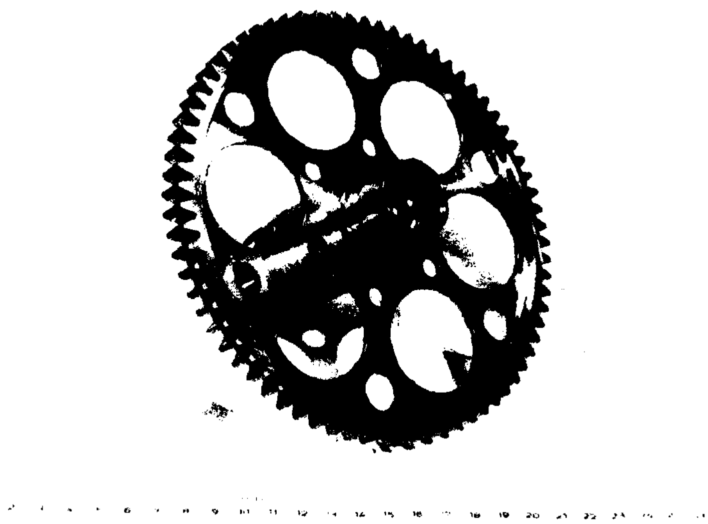

FIG. 5: HYDRAULIC PUMP GEAR

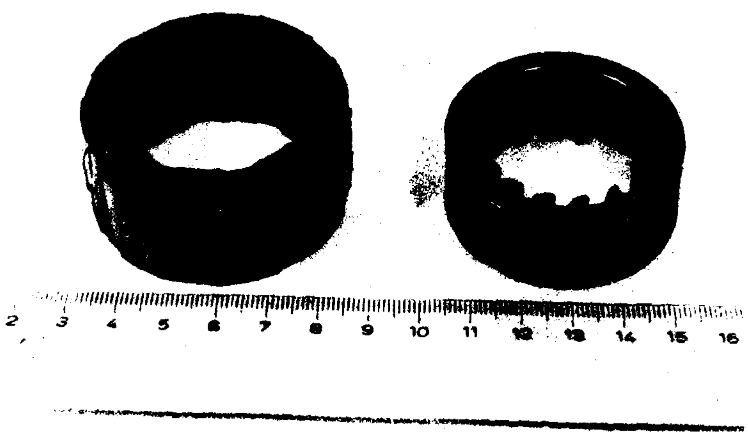

FIG. 6: FREEWHEEL CLUTCHES OF SPS GEARBOX

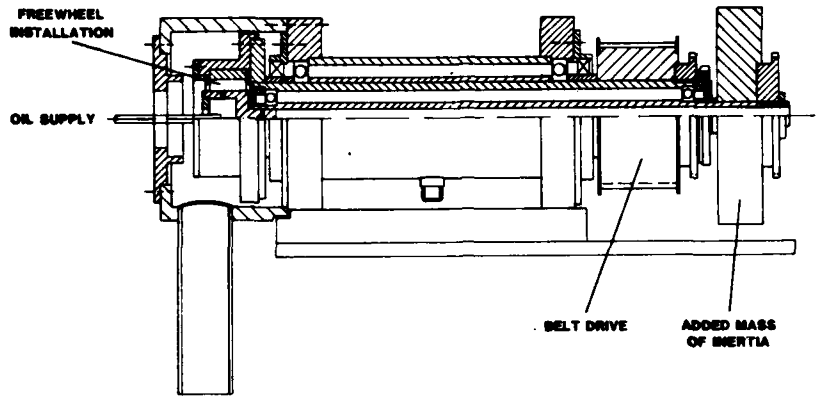

FIG. 7: TEST RIG FOR INVESTIGATION OF THE ENGAGING BEHAVIOUR OF FREEWHEEL CLUTCHES

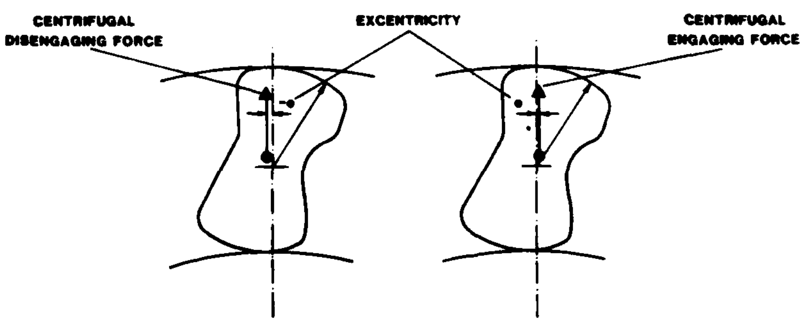

FIG. 8: SPRAGS WITH DIFFERENT CENTRIFUGAL BEHAVIOUR

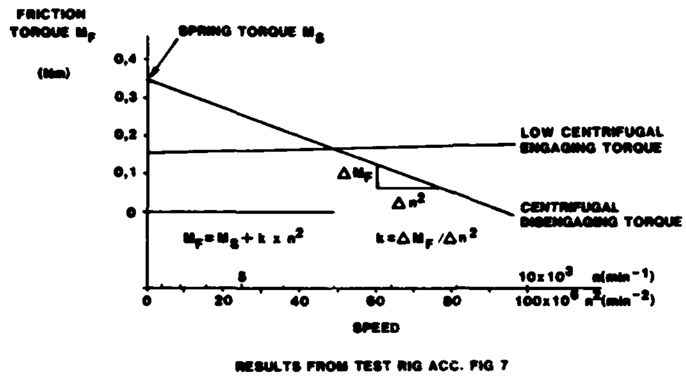

FIG. 9: FRICTION TORQUE OF DIFFERENT FREEWHEEL CLUTCHES

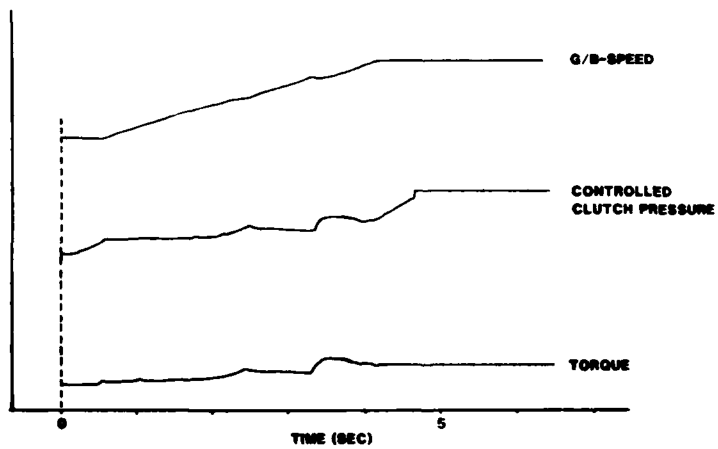

FIG. 10: TYPICAL CHARACTERISTIC OF CLUTCH ACCELERATION CONTROL

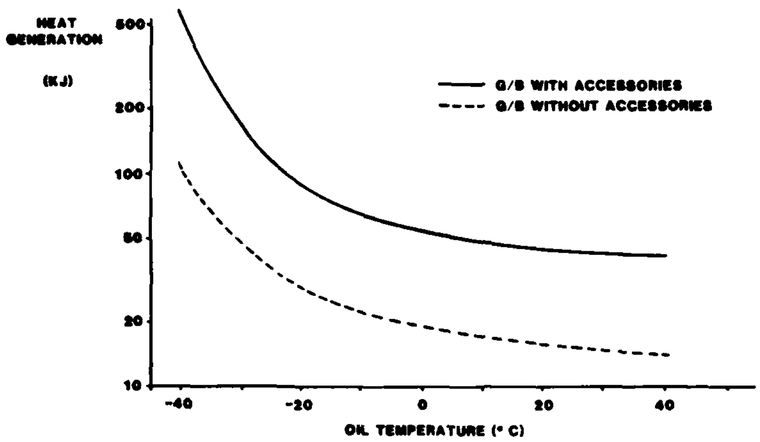

FIG. 11: HEAT GENERATION WITHIN THE APU CLUTCH DURING ENGAGEMENT

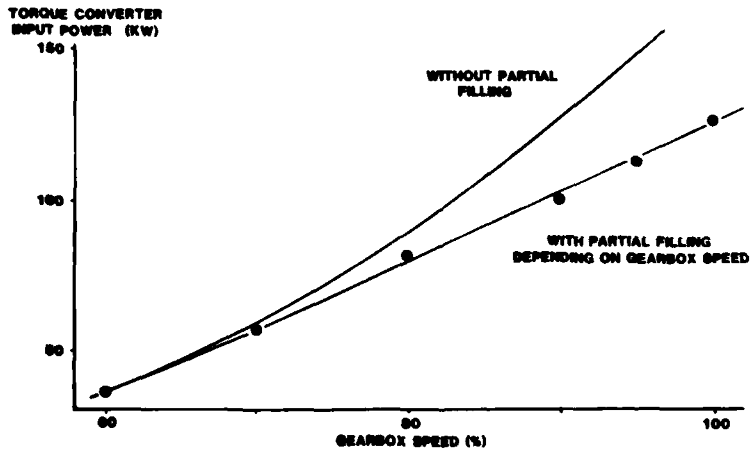

FIG. 12: POWER REDUCTION OF TORQUE CONVERTER BY PARTIAL FILLING

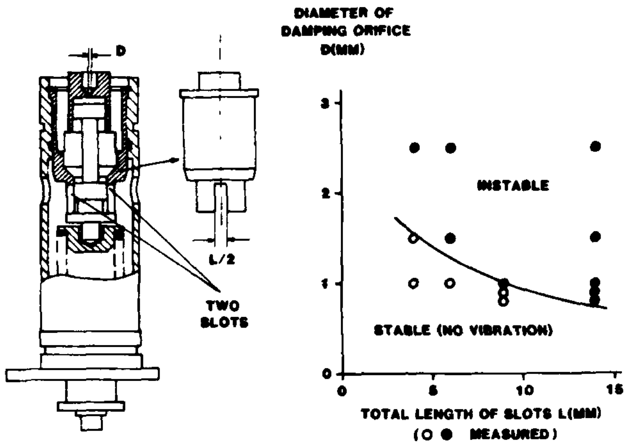

FIG. 13: HYDRAULIC VIBRATION CHARACTERISTIC OF CLUTCH PRESSURE RELIEF VALVE

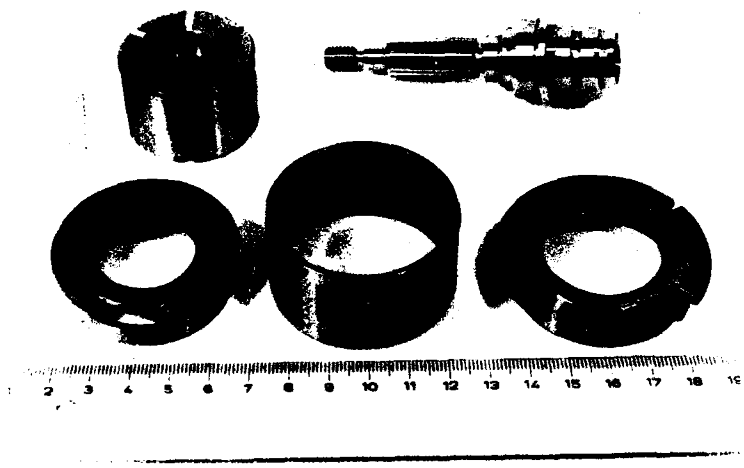

FIG. 14: VANE PUMP FOR GEARBOX PRESSURIZATION

AD-A168 241 CONFERENCE PROCEEDINGS: AIRCRAFT GEAR AND BEARING TRIBOLOGICAL SYSTEMS HE..(U) ADVISORY GROUP FOR AEROSPACE RESEARCH AND DEVELOPMENT NEUILLY.. FEB 86 2/3

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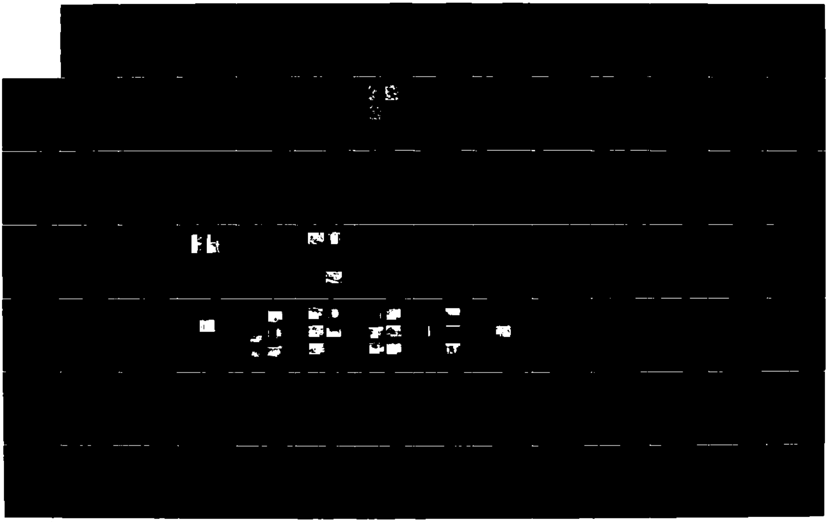

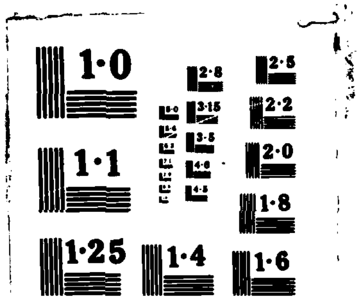
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Specification Clause	O - 160						O - 156*
	Requirements DERD 2497	Test Method ASTM	Test Method IP	Esso ETO 25	Shell Asto 555	Castrol 599	Castrol 5000
Specific Gravity at 15,6/15,6 °C	Report	1298	160	0,995	0,994	0,995	0,975
Total Acid Number mg KOH/g	Report	664	177	0,4	0,3	0,2	0,03
Viscosity at 204 °C: cSt	1,3 min	D 445	71			1,43	1,46
Viscosity at 99 °C: cSt	5,5 max	D 445	71	5,25	5,37	5,28	5,04
Viscosity at 37,8 °C: cSt	25,0 min	D 445	71	28	29,25	29,3	26,2
Viscosity at - 40 °C: cSt	13000 max	D 445	71	9500	10560	12500	7640
Pour Point: °C	- 54 max	97	15	- 56,6	- 59	- 57	- 60
Load Carrying Ability IAE Gear Machine							
2000 rpm (- reference, RR.1)	100	-	166	185	156	123	83
6000 rpm (- reference, RR.1)	100			165	168	173	106

*) O - 156 Lubrication oils are specified acc. MIL-L-23699 C

FIG. 15: COMPARATIVE DATA OF GAS TURBINE ENGINE LUBRICANTS

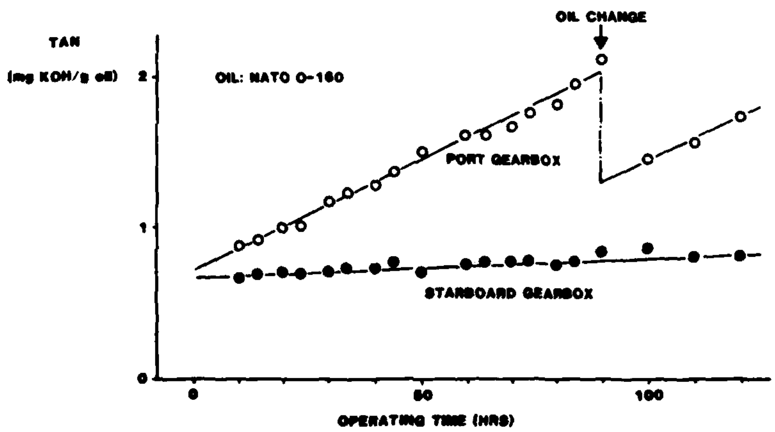

FIG. 16: ACIDIFICATION OF GEARBOX LUBRICATION OIL DURING RIG ENDURANCE TEST

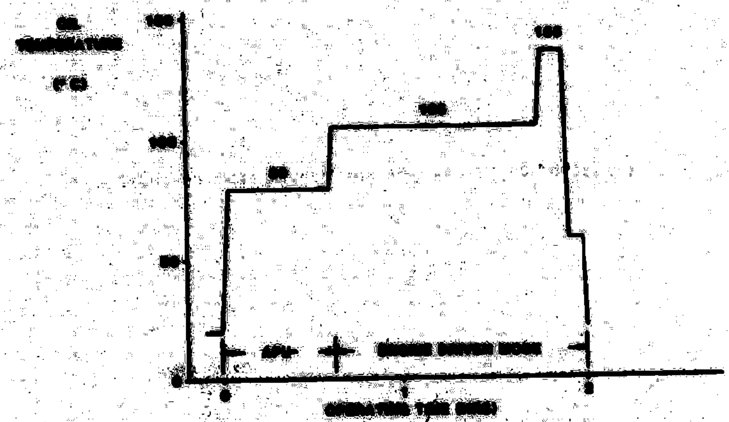

FIG. 17: TEMPERATURE PROFILE FOR THE ENDURANCE TEST CYCLE

Requirements on Lubrication Oil from the View of a Helicopter Manufacturer

by

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ABSTRACT

Experience with known oil types, used in helicopters for transmissions, engines, hydraulics and brakes has shown, that 3 main characteristics if available, would improve drastically the performance of the aircraft.

These 3 main characteristics are:

- The same type of oil should be usable for all the above mentioned systems.
- The oils brands, produced according to a common specification, but by different manufacturers, should be mixable without restrictions.
- Some features of the oil, which are specially important for helicopter operations, should be improved.

INTRODUCTION

In accordance with the technical state of the art in helicopters oils or fluids are used. These oils and fluids have optimized properties for their applications in systems such as engines, transmissions, hydraulics, landing gear etc. The manufacturers of these oils and fluids are doing a lot of effort to improve their products. The helicopter manufacturers are incorporating the improved product qualities to guarantee lower operating costs.
Normally improved oil qualities are required, so that the performance ratings can be increased, life time of components can be extended, or that the operational limits can be enlarged without a redesign of the affected components in an existing helicopter.

When ever possible, the helicopter manufacturer endeavors to minimize the quantity of different oils and fluids. For military helicopters, an even harder requirement for a reduction of different oils and fluids have to be matched. The resulting disadvantages within the according systems as long as they are not too strong are accepted. Logistical requirements, especially flight safety aspects caused by unintentionable incorrect maintenance actions are the main reasons to reduce the number of oils as far as possible.

Looking to existing helicopter components such as transmissions, turbines and piston engines, the one oil solution has been established. For other equipments, similar requirements exists. Nevertheless oils with more appropriate characteristics for transmissions are available.

For the future we therefore expect logistical requirements, because the stockceeping of the oil, the time between oil changes and the life time of the components are the important parts of the maintenance costs. Beside these logistical requirements, technical and flight safety aspects become more important. These requirements concern the reliability of the oils and the possibilities of errors which can happen during necessary maintenance actions.

The maintenance errors normally have one common reason:
The parted or complete incompatibility of different oils and fluids when they are mixed. It should be a requirement that all oils and fluids which are produced in accordance to the same specification should be mixable without any restrictions. This means that after mixing, the specified values of the relevant parameters are guaranteed.

In existing specification requirements for possible mixing such as oil MIL-L-23699 with MIL-L-7808 are listed, but there are no requirements that permit the mixture of oils produced by different manufacturer but with the same specifications. Examining existing specifications it would be very difficult and expensive to perform all the possible mixings and control the single specified values. For new specifications this should be a strong requirement, where even lower values which would automatically mean lower operating conditions could be accepted.

Currently the helicopter manufacturer must decide whether to use special oils which have to be tested for civil applications or use available oils which are qualified.

It is quite clear, that this procedure ist extremely expensive and time consuming. Therefore such necessary qualification tests are to a certain extend performed together with the helicopter customer. The disadvantage of such procedure is, that these tests cannot represent completely the certification limits because power and temperature limits will be reached accidentally. But even if all oils are qualified for use, the question of mixing these oils is not answered.

Nearly all oil producers recommend that it is safer not to mix oils with the same specification number but produced by different manufacturers. The question should be allowed why this recommendation is declared. But on the other hand, oils with different specification numbers are allowed to be mixed. We assume that the compatibility of two different oils is depending on the fact, that both oils are built up with the same or at least similar basic substance.

If we follow the recommendation of the oil producer, it is necessary in any case that a complete oil exchange has to be performed if the same oil brand is not available.

In contrast to the above mentioned procedures and recommendations, is the situation on the field of helicopters in military use.
All oils which can be used are marked by a special code number combined with the specification number. The oil producer can only be identified if the code number is known. Therefore in practice oil with the same specification number but different manufacturer code numbers are mixed. To minimize the risk, the helicopter manufacturer is running extensive tests but the military customer is accepting this risk. To minimize these risks, the number of oil manufacturers is kept as small as possible. Due to the fact, that military customer are the main consumers, this market is highly competitive. The produced oils only fulfill the minimun requirements so that the lowest cost can be offered. This could be verified by company tests where oils MIL-L-23699 and NATO-0-156, which are produced by the same manufacturer, were compared.

Oil is an important factor for the operation and flight safety of the helicopter. Oil characteristics influence the design from the beginning, are a major factor in the direct operating cost (DOC) and can easily cause operating troubles by improper handling.

Therefore we recommend that the existing situation has to be changed. To eliminate or at least minimize of the human factor aspect for higher flight safety operations with lower DOC's, the characteristics of oils must be improved.

EXPERIENCE WITH EXISTING OIL SYSTEMS

Transmissions

Today only oils specified within MIL-L-23699 are used from the beginning of the design. Even for extreme operating conditions such as -40°C to +50°C they have sufficient characteristics to allow a proper design.
For special request of operators, which are mainly flying under arctic conditions, oils specified within MIL-L-7808 were certified. This was done for logistical reasons because engines are only allowed to be operated with this oil, if temperatures are below approximately -30°C.
All oils specified for MIL-L-23699 as well as MIL-L-7808 do have a better load carrying ability as required. By extensive tests (FZG-test A 8,3/90 per DIN 51354) with a special FZG test equipment, the oil with the lowest values in relation to load carrying capability was investigated (see table 1).
With these tests it was guaranteed that all development and certification tests would cover the worst case.
The values for the viscocity of the oils were within the given limits. In addition to the tested values Ryder Gear test results, which were determined by the oil vendors, are listed in table 1.
Together with our operators the different oil brands have demonstrated step by step their capabilities within the scheduled time between oil changes. This procedure is highly time consuming and causes troubles because immediately after certification, only a few oil brands are certified. It would be a great advantage, if there would be no necessity during the qualification to demonstrate each oil brand over the total time between oil changes. To allow this, the certification authorities should agree to a new test procedure.
Different oil vendors were asked to clarify the question of mixing oil produced under the same specification. Their answers normally were that it would be better not to mix. They are based on the fact that mixing is not a requirement which is necessary to certify the product and that therefore no actions have been performed.

It could be a guarantee issue, which is yet not been proven, that the helicopter manufacturer cannot assure such guarantees.
If the oil vendors permit the mixture of different oil brands, it is due to the fact, that these oils consists on the same basis, mixed up with the same additives but different mixture ratio which are on the market under different trade names.

Our experience with test results, where we mixed different oils, has demonstrated that no negative chemical reaction, which changes drastically the quality of the oil up to a temperature of +120°C, occured. It could also be shown that the relevant

properties important for the transmission such as scuffing tendency, viscosity and thermal conductivity factor also were not degraded for the mixed oils. Colour changes were observed even by very small mixture ratios.
Our opinion to this subject is, that only an amendment to the specification will help to improve the situation. For oils, which are used in transmissions, it is our experience that it is not necessary to demonstrate all properties over the total temperature range, but is is mandatory that such restrictions are published. It could also be discussed, that mixed oils are allowed to have certified deviations from the minimum values of the specifications. Up to now our experience with tests, as well as with military helicopters in service, have shown that mixing of oils from different brands will have no catastrophic consequences. But these experiences are not sufficient for a proof. The situation has to be clarified by common rules.

Beside the possibilities of mixing oil MIL-L-23299 with oil MIL-L-7808 we could demonstrate that hydraulic fluid MIL-H-5606 and oil MIL-L-23299 are mixable.
When the hydraulic pumps are directly driven by the main transmission, hydraulic fluid can enter the main gear box in case of undetected failures of seals for example.
To understand what happens under these circumstances, test were conducted to substantiate the Failure Mode Effect of the transmission. The most important factor in this case was the test results shown in table 2, which show that there is a nearly linear behaviour of the scuffing tendency to the mixture ratio.
Further differences such as a chemical change could not be observed. This example also shows that other mixtures beside MIL-L-23699 oil with MIL-L-7808 oil should be considered as a further example. Similar tests with MIL-H-83282 (Hydrocarbon) have shown that FZG load stages of 8-9 could be reached, which gives new possibilities in the direction of possible mixtures and might be applications.

Oil vendor	F 26 Load stages MIL-L-23699 (commercial)	F 26 Load stages 0 - 156 (military)	Ryder gear test data
A	7	7	2500
B	8 to 9	7	-
C	7 to 8	7 to 8	2750
D	7 to 9	n.a. *	2800
E	7	7	-
F	9	n.a. *	3200
G	9	9	3100
H	9	n.a. *	3200
I	8	n.a. *	3100

* n.a. = not available

Table 1: Oils per MIL-L-23699 in FZG test
(A 8.3/90 per DIN 51354)

Step	Oil quantities MIL-L-23699 %	MIL-H-5606 %	FZG load stage
1	100	0	9
2	90	10	8
3	80	20	7
4	70	30	6
5	60	40	5
6	50	50	4
7	40	60	3
8	30	70	-
9	20	80	2-3
10	10	90	2
11	0	100	2

Table 2: Scuffing tendency of different oil mixture ratios by tests
(A 8.3/90 per DIN 51354)

Hydraulics

In the same way as at the transmissions, which had been tested with hydraulic oils, the effect of transmission oils tested with hydraulics have been studied.

Hydraulic tests have been conducted in two steps.

In the first step a hydraulic system which operates with mechanical control spools was tested.
In the second step an electrical servo valve was added. The tests were conducted at environmental conditions of -30°C to +70°C. The test hydraulic was a normal servo control with a pressure supply unit and without any modifications, as used in a helicopter.
It was determined while starting at low temperatures, that the function of the hydraulic was effected by the used hydraulic fluid (oil according to MIL-L-23699) according to its different viscosity temperature behaviour.
A preliminary heating of the hydraulic fluid at the starting phase was only necessary below temperatures of -10°C. A restriction of the hydraulic-functions was determined at the test at temperatures below +5°C when the electrical servo valves were fitted. In this case the effect of the viscosity and of the contamination of the oil was substantial.Wear at the control spools, piston or at the pumps could not be noticed. A reason might be due to the limited test time of about 100 hours.
It can be conclusioned that oils according to MIL-L-23699 are usable as hydraulic fluid in principle and that they are used in special cases for example at engines.
A servo hydraulic with pressure supply unit, which shall be used for helicopter control, has to be modified according to the data of the used fluid. As a consequence, all components belonging to the hydraulics (as pumps, reservoirs, valves, sealings and so on) had to be designed and qualified new.

The functional problems at low temperatures can be solved by using a heating system. But the weight and cost of such a system is not attractive for a helicopter, therefore it is not practicable.
An admixture of additives to get a lower viscosity has to be eliminated because of logistical reasons.
A simple oil change from MIL-H-5605 to MIL-L-23699 is not a solution for the considered temperature range. Using oil according to MIL-L-7808 the situation will be better but it isn't a solution as well.

So MBB is not looking any longer at the use of transmission oils as hydraulic fluid for helicopter control.

Looking at the logistical side of the use of transmission oils at hydraulic systems, all arguments are against this use. One of the severest reasons is, that the hydraulic components will be tested at universal testbenches. These testbenches will be used for hydraulic systems of several different airplanes and helicopters and must be adaptable or new testbenches must be provided according to the kind of oil used.

In this case failures might occur which could be hazardous when working with ground support equipment that has hydraulics mounted to helicopters.
As a rule, modern hydraulics were designed as a closed system which are supplied only by ground support eqipment. So the risk to mix up the oils is minimized because of the geometry of the outboard couplings.

So MBB is only looking at the risk of mixing the oils of engines and transmission of their helicopters.

Engines

The statements made under para Transmissions are in a similar way valid for engines. The application of oils in engines will render more difficulties than in transmissions, since the oil is operated in an engine at a much higher temperature.
For this case, the problem of substantiation has been transfered to the engine manufacturer.
Since the engines have an oil consumption due to their conception, oil has to be replenished as needed. For this point the problem of miscibility will be intensified when the helicopter operator will use his helicopter on several bases.

For civil application, most engine manufacturers prescribe the allowable oil types to the helicopter manufacturer and the helicopter operator in their operating manuals; including the requirement not to mix different oil types. Some engine manufacturers only give the corresponding specifications and simultaneously require that the substantiation for the individual oil types has to be performed by the helicopter manufacturer or by the helicopter operator at his own risk.

For performing the substantiation test, a corresponding documentation will be supplied by the engine manufacturer. The expense and the risk will remain. To minimize the expense and risk for the individual oil types, the tests are performed without mixing.

Oil types which are designated as miscible by the engine manufacturer are normally produced by the same oil manufacturer and are sold with different trade names. Tests with oil types, which were, according to the engine manufacturer not miscible, have shown that no severe problems occur during a complete oil change period.
Only an increased carbon formation which was within the allowable limits was ascertained. A change of acid content in the oil was not ascertained.
For engines which are used for military purpose the situation is as vague as for the civil application.

For military application it is easy to get clearification because the specifications are stated by the authority, the army is the greatest oil consumer and sufficient labs and test facilities are available.

OIL REQUIREMENTS

If we take the recent experience which was shortly discussed in the previous chapters as a basis the following suggestions and requirements are claimed to improve existing and future oils.
These requirements do not content any chemical discourse. This should be done by the experts in that field.
To restrict the statements only the oils MIL-L-7808, MIL-L-23699 and MIL-H-5606 are considered, but the requirements are also relevant for other oils and fluids.
We assume that MIL-L-7808 and MIL-L-23699 will be the basis for further developments. They are introduced world wide and have proven their quality. The situation with MIL-H-5606 oil is that it will be replaced step by step for example with MIL-H-83282. The new oil MIL-H-83282 is fully exchangable with MIL-H-5606, but there are some problems at very low temperatures with this oil which have to be investigated.

The trend to develop helicopter components with higher realiability, more performance, longer useful life, wider operating conditions, etc. continues.
New and special oils are necessary. These oils will be optimized for special applications, but can be contrary to logistical and flight safety requirements caused by human fault behaviour.
We propose that just these requirements should be the most important ones when such oils are used at in series produced helicopters.

A. Requirements for transmission and engine oils:

1. Miscibility of oils which are produced with the same specification (If necessary even operating limitations could be accepted.)
2. Specification of the load carrying ability (Ryder gear test or FZG-test) to allow a direct use of the test data, instead of inaccurate percentage calculation of reference oils presented as function of the temperature.
3. Specification of the specific heat as a function of the oil temperature. Requirements under 2. and 3 are important parameters for an optimized design of transmissions and oil cooling system, particulary if volume and total weight (including necessary oil quantities) are main decision criteria.
4. Limitation of the oil viscosity for low temperatures to a value of approximately 7000 ÷ 10000 centistokes at -40°C, and simulateous to a value of higher than 7 centistokes at +100°C. (At operating conditions of +130°C a minimum viscosity of 3 centistokes should be available).
5. Improvement of the corrosion protection ability to an extend, that special preservation oils can be eliminated.
6. It should be investigated if the qualification tests on transmissions could be performed with a special reference oil to avoid the necessity of testing all oil brands which are on the market. This reference oil could be either a special oil or a mixture from different oil brands.
   Depending on the results of necessary tests to establish this procedure on transmissions it should be decided if this could be also used for engine qualifications.

B. Requirements on hydraulic fluids

Independent from special hydraulic specific requirements to the standard oil MIL-H-5606 and the new MIL-H-83282, additional information is necessary. They are important if those oils are intentionally or unintentionally used in transmissions.

1. The load carrying ability should be specified.
2. The values for specific heat capability should be specified.

CONCLUSION

The specifications for oils do not explicitly allow the mixture of oils produced with the same specifications but by different manufacturers. This requirement should be obvious, it is not certified.
Tests were run in our house to investigate the effects of oil-mixtures. These tests have shown that no catastrophic or even negative effects, other than expected, were experienced.
A general conclusion cannot derived only by these test results. It might be difficult to clean up this situation with the oils now on the market, but it should be a must for all new developments and it should be forced by changing the specifications respectively.

In addition the specification should require to establish data, which are necessary to design new improved transmissions.
To achieve a better situation, it is necessary that all partners, beginning with authorities, laboratories, manufacturers, procurement officies, helicopter and engine manufacturer and the customer have to work much closer together, to reach the target.
The helicopter must be safer and more reliable. The human fault behaviour should be considered from the beginning of the design. If not so, the origin of possible failures is created and will destroy the best design.

DIRECTION OF R&D AND CURRENT STATUS OF UNDERSTANDING OF ADVANCED GEAR STEELS

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ABSTRACT

High performance gears in the modern helicopter must operate at ever increasing torque and RPM. A major consequence of this increased torque and RPM is a significant increase in the surface temperature and increased scuffing of the gears. In response to this problem, Boeing-Vertol introduced a new class of steel (hot work tool steel) as a high performance aircraft gear steel and started a trend which all subsequent work has confirmed to be a correct decision. This report focuses on the three most prominent candidate critical high temperature aircraft gear steels including Vasco X-2M, CARTECH X-53 (PYROWEAR 53), and CBS600. The heat treatment responses of these alloys will be compared. Three additional alloys (M50NiL, CBS1000M, and AMAX B) will be discussed in less detail. Most of the information contained in this report is metallurgical in nature with the appreciation that considerable engineering and manufacturing information is available for most of the alloys, but with the realization that much work still needs to be done.

INTRODUCTION

The modern helicopter is continually being required to lift ever heavier loads and to do this at ever increasing speed. The net effect of this is that the gears which carry this power from the turbine to the rotor are subjected to higher torques at higher RPM. An important consequence of this increased torque and RPM is that a significant increase in temperature at the contact surface of the gear can be expected which can only be partially alleviated by more robust lubrication cooling systems. Estimates as to the expected surface temperature range from 200 to 300C. This surface temperature should be distinguished from the oil-out temperature which has a maximum of about 120C. This oil-out temperature does not fluctuate widely during flight from the 'standard' operating temperatures and is continuously maintained by control of the oil-in temperature by a lubrication cooling system.

The workhorse of the major American helicopter producers for critical high performance gearing is the carburizing grade steel AISI 9310 (Table 1). This steel is only recommended for continuous use at temperatures up to about 150C, above which it will experience serious losses in strength with long exposure times. Because of this loss in elevated temperature strength, the scuffing resistence of AISI 9310 becomes increasingly suspect in the new environment of increased torque and RPM (and estimated surface temperatures of 200-300C). The evidence for this deteriorating scuffing resistance is difficult to document since the rejection of the great majority of gears at Army rework facilities are based on very subjective rejection criteria. The vast majority of gears are rejected because of pitting but no effort is made to determine if the pitting is from scuffing or from fatigue effects. The differentiation between these is difficult for in-service failures and should be carried out in a laboratory rather than shop environment.

In the late 60's, Boeing-Vertol (B-V) decided that a replacement for AISI 9310 was highly desireable for advanced drive systems. They no doubt arrived at this conclusion based on observations of gears taken out of service, good engineering, and at least some leap in faith. In an engineering study comparing several steels including VASCO X-2, Bower 315 etc., Vasco X-2 was selected as having the best combination of properties necessary to produce good aircraft quality high performance gears. The one area where Vasco X-2 was clearly at a disadvantage (compared to AISI 9310) was in the values of fracture toughness. This of course was not too surprizing since it has long been appreciated that alloy steels (such as AISI 9310) have higher fracture toughness values than tool steels (Vasco X-2 of course is a low carbon variation of H-12 which is a hot work tool steel). The fracture toughness of AISI 9310 in fact is so high (above 100 M Pa$\sqrt{m}$) that it is not considered a critical design parameter. With the use of Vasco X-2 the situation changed and fracture toughness became a very important consideration if not a design parameter. In fact, the fracture toughness has perhaps assumed more importance than is justified, since the required fracture toughness for high performance gear applications

is not known. A general concept has generally been accepted, however, that fracture toughness shall be as high as possible consistent with meeting other property requirements such as strength, fatigue resistance, and ductility. It should be emphasized that while the gear surface must be hard and wear resistant, the webbing of the gear (the low carbon core material) must be resistant to crack initiation and propogation. In fact Boeing-Vertol employs the threshold (as determined from crack progagation studies) of the core material as an important parameter in the design of the web of the gear. While the fracture of the carburized tooth is certainly not desireable, it does not present the possibility of instant disaster of a fracture which has initiated in the web of the gear.

The introduction, therefore, of Vasco X-2 as a high performance aircraft gear steel by Boeing-Vertol started a trend which all subsequent work has confirmed to be a correct decision. The following report will focus on the three most prominent candidate steels including Vasco X-2M (a lower carbon version of the original Vasco X-2), CARTECH X-53 (PYROWEAR 53), and CBS600. The heat treatment responses of these alloys including hardness, tensile properties, fracture toughness, microstructure, and Charpy energy will be presented. Three additional alloys will be discussed in less detail but with specific advantages not present in the other alloys such as ease of carburizing, conservation of alloying elements, or reduced susceptibility to embrittlement during aging. Vasco X-2M (VIMVAR) is the only advanced high temperature carburizing grade steel currently in use today in critical gear applications in aircraft. This does not mean that it is the only steel that could be used for this application but it does mean that considerable engineering and manufacturing effort has gone into making this a successful high temperature (high hot hardness) high performance gear steel. Any successor to VASCO X-2M should not merely be marginally better but should have some property or processing characteristics which are clearly superior or should eliminate a weakness in VASCO X-2M that could be critical to the long range surviveability of the gear.

The present report compares the properties of alternative steels to those for VASCO X-2M. The three (3) initial candidate steels have been evaluated in considerable engineering detail by BOEING-VERTOL (VASCO X-2M), Bell (CARTECH X-53) and Sikorsky (CBS600). Most of the information, however, contained in this report will be of a metallurgical nature. The report will present data on VASCO X-2M in considerable detail while the data on the other alloys will be presented to illustrate the essential differences. As will be realized all the facts are not in and we will attempt to point out where we feel more work is needed.

RESULTS AND DISCUSSION

Figure 1 shows the effect of tempering temperature on the R_C hardness values for the initial three (3) candidate high temperature carburizing grade steels in comparison to AISI 9310. Table 1 shows the compositions of these alloys. The essential differences between these alloys which reflects the compositions will be discussed. The AISI 9310 curve is typical of high strength alloy steels which show continuous softening for tempering temperatures above about 150C. The hardness values of CBS600 also drop off continuously with increasing tempering temperature but the rate of softening is retarded. Vasco X-2M and CARTECH X-53 are typical of the so-called secondary hardening steels where a secondary hardening peak occurs at about 500C. These steels are much more resistant to softening. It is estimated that VASCO X-2M, CARTECH X-53, and CBS600 can operate continuously at 315, 275, and 230C respectively. Also shown in Fig. 1 are the maximum temperature ranges for present and future helicopter transmissions.

VASCO X-2M

Fig. 2 shows the effect of austenitizing temperature on the R_C hardness of 0.15, 0.24 and 0.35 w/o carbon VASCO X-2 where 0.35 carbon is the hot work tool steel H-12. The increase in hardness is principally due to the decrease in the amount of primary ferrite with increased austenitizing temperature as is shown in Fig. 3 (0.15C and 0.24C) and metallographically for Vasco X-2M in Fig. 4. While the amount of retained austenite varies between 3 and 5% with increased austenitizing temperature, the effect of these small changes on the hardness is small.

Fig. 5 shows the effect of tempering temperature on the hardness for various austenitizing temperatures. A secondary hardening peak is observed at about 500C. A slight increase in hardness is observed for austenitizing temperatures above 1010C which is the commercially used hardening temperature. This increase is no doubt due to the increased resolution of the alloy carbides at the higher temperature.

Fig. 6 shows the effect of tempering temperatures on the K_Q fracture toughness for various austenitizing temperatures. For the commercial heat treatment of 1010C followed by a 315C temper, the K_Q is about 45 MPa$\sqrt{m}$. By increasing the austenitizing temperature to about 1125C, the fracture toughness can be significantly increased to over 60 MPa$\sqrt{m}$) but the grain size becomes very large for austenitizing temperature above 1100C. At a tempering temperature of about 500C, there is a complete disappearance of a shear lip on the fracture surface. This observation is consistant with the drop in fracture toughness for tempering temperatures above 315C. The values of fracture toughness reported here are for vacuum arc remelted (VAR) material. This value has been increased to about 70 MPa$\sqrt{m}$ in vacuum induction melted vacuum arc remelted (VIM-VAR) material which is in current usage by Boeing-Vertol in the CH 47-D. The carburizing of VASCO X-2M is difficult because of the high chromium content. Uniform carburizing, however, can be obtained by a pre-oxidizing treatment before the standard endothermic gas carburizing or

by vacuum carburizing. (Vacuum carburizing is a process which uses no carrier gas, can be carried out at high temperature, and eliminates surface oxidation.) A program is currently underway to qualify the vacuum carburizing process for critical aircraft gear applications.

The effect of tempering temperature on the Charpy energy of Vasco X-2M is shown in Fig. 7. Also shown in this figure is the effect of 1000 hours aging at 260°C. The decrease in Charpy energy for long term aging at 230C has considerable practical significance 260C has same metallurgical significance. In the short term, the major helicopter producers agree that the effect of long term aging at 230C is related to the expected increased surface temperature of main drive transmission gears under high torque and RPM conditions. In the long term, of course, it is expected that transmissions will operate at temperatures (oil-out temperature) exceeding 200C. The effect of long time exposure at temperatures of 260 and 230C on the Charpy energy has been investigated in considerable detail for CARTECH X-53 and will be discussed below:

Another important requirement for high temperature gears is that the hardness shall be unaffected by long term exposure at elevated temperature. Table 2 shows the effect of aging 1000 hours at 315C on the hardness (Ref 4). AISI 9310 softens considerably with long term aging while Vasco X-2M retains its hardness even after extended aging.

The fracture toughness in carburized cases for many of the same steels is shown in Table 3 (Ref 4). Again the fracture toughness was determined before and after exposure for 1000 hours at 315C. It can be seen that the toughness decreased about 50% after exposure for Vasco X-2M. It must be emphasized here that Vasco X-2M is not currently used above about 125C and that the method employed for measuring the fracture toughness of a carburized case is not an ASTM standard test procedure. The change in the fracture toughness, however is felt to be significant and the effect of long time aging at 230C of the core (low carbon) Vasco X-2M is currently being investigated in much more detail.

CARTECH X-53 (X-53)

The effect of austenitizing temperature on the properties and microstructure of X-53 closely parallels those of VASCO X-2M. A major difference is the absence of even the small amount of retained austenite observed in Vasco X-2M. The commercial austenitizing (hardening) temperature for X-53 is 910C vs 1010C for Vasco X-2. With tempering, X-53 has a secondary hardening peak at about 500C. The Charpy energy decreases sharply and continuously for tempering temperatures above 260C (Fig. 8). This decrease is independent of austenitizing temperature between 850 and 950°C. Included on the same figure is the effect of 1000 hour aging at 260F on the Charpy energy. The large drop in Charpy energy with long time aging at 260C is of considerable practical significance. First of all, X-53 is a strong candidate as a high temperature high performance carburizing grade gear steel because of its high fracture toughness. Because of the critical nature of the observation and the feeling by representatives from the four (4) major U.S. helicopter producers that 230C (450F) would be a more appropriate aging temperature, the data in Fig. 9 was collected. The samples used in the production of the data in Fig. 9 were heat treated at Bell Helicopter according to their standard specifications including a pseudo-carburizing treatment. Aging treatments at 230C as a function of time and the Charpy impact testing were carried out at AMMRC. Each point on the curve represents the average of three (3) readings. The material employed to collect the data in Fig. 9 is double vacuum melted (VIM-VAR) while the material employed to collect the data in Fig. 8 was air melted plus VAR. There is more scatter in the data than would normally be expected for Charpy energy determinations. This scatter appears to decrease for longer aging times. The mechanism for embrittlement has been shown (Ref 5) to be the precipitation of alloy carbides (M_6C) at prior austenite grain boundaries. The scatter in the Charpy energy data for aging times up to 1000 hrs at 230C is not understood and there is no reason to suspect either the Bell or the AMMRC heat treatments.

The loss in Charpy energy appears to be fully recovered for aging times of 2000 hrs at 230C and the scatter in the data is much reduced. Since the hardness is not effected by this long aging treatment, it suggests measures that can be taken to eliminate both the scatter and more importantly the drop off in Charpy energy (as yet subject to considerable scatter) for long time low temperature aging.

One obvious solution to the drop in toughness would be to duplicate the 2000 hrs. at 230C aging treatment by a significantly shorter aging (tempering) treatment at 315C. Since the activation energy for the diffusion of molybdenum in iron is 57.7 kcal, the time to attain equivalent diffusion at 315C is only about 0.67 hours (see Table 4). But this calculation is based on substitutional diffusion and we know that grain boundary and dislocation diffusion will predominate at the low aging temperatures being considered. If we assume an activation energy of 30 kcal for the low aging temperatures, we calculate a time of about 30 hrs. at 315C to be equivalent to 2000 hrs. at 230C. Since the region near the prior austenite grain boundaries are observed to be devoid of carbides, a combination of diffusion mechanisms is felt to be operative. In summary, if specimens are tempered at 315C for 30 hours, embrittlement of our structure should be avoided with no loss in strength and subsequent in service aging at and below 315C should have no effect on the toughness. An experiment testing this idea is currently in progress at our laboratory.

CBS600

CBS600 was developed by Timken as a carburizing grade bearing steel for service above 150C. The philosophy used in designing the alloy was to retard the decomposition of the martensite by additions of silicon, chromium, and molybdenum. This was quite different from the philosophy used in the development of secondary hardening steels (Vasco X-2M and CARTECH X-53) which achieve their high temperature properties by the development of stable alloy carbides of molybdenum, vanadium, and tungsten. CBS600 carburizes much more readily than for example VASCO X-2M, presumeably because of the lower chromium content (1.5 W/O vs 5.0 W/O for Vascos X-2M). Jatczak (Timken) feels that CBS600 should be capable of operating at a maximum service temperature of 450F (230C).

The effect of austenitizing temperature on the mechanical properties and microstructures of CBS600 closely parallels those for Vasco X-2M and X-53. The commercial austenitizing (hardening) temperature for CBS600 is about 910°C which is just about at the beginning of the plateau region of the hardness vs austenitizing temperature curve. The effect of temering temperature on the hardness for several austenitizing temperatures is shown in Fig. 10. No secondary hardening peak is observed for any austenitizing temperature but the hardness retention with increased tempering temperature is considerably better than for AISI 9310 steel (Fig. 1). Fig. 11 shows a composite of the effect of tempering temperature for specimens austenitized at 900C on the tensile, Charpy energy, and K_Q fracture toughness. The ultimate tensile strength, and Charpy energy all decrease continuously if not sharply with increased tempering temperature while the yield strength and the K_Q fracture toughness do not change significantly over the same range of tempering temperature. The sharp drop in Charpy energy with increased tempering temperature of X-53 is not observed for CBS600. The values of Charpy energy and K_Q fracture toughness are representative of single vacuum melted steel and would be higher for double vacuum melted (VIM-VAR) steels. The effect of long time aging at 230C on the Charpy energy for CBS600 is currently underway at AMMRC.

M50 NiL

General Electric, in cooperation with Massachusetts Institute of Technology and Fafnir Bearing Company under contract to the Air Force and Navy is performing a program designed to identify a material suitable for ultra-high-speed rolling element bearing operation (up to 3 million DN). In addition to having rolling contact fatigue life and hot hardness as high as for M50 (the high hot hardness bearing steel widely used in aircraft gas turbine engines in the USA) the new material must have a high fracture toughness. M50 NiL was the material developed on this program as a carburizing grade rolling element bearing.

The evolution of M50 NiL began as a low carbon modification of M50. The reduced carbon level (0.15% versus 0.80%) provided the potential for surface hardening by carburizing thus creating a bi-hardness structure. The addition of 3% nickel was to provide ferrite control and stability, added toughness, and improved fabricability. M50NiL's success as a bearing steel, comparable to homogeneous M50 steel with much improved fracture toughness, suggests that it should be applicable as a carburizing grade high temperature gear steel. A major drawback may be its high alloy content which may increase the cost of materials to unacceptable levels. Preliminary vacuum carburizing tests at AMMRC indicate that it readily carburizes at about 1050C in spite of the 4.5% chromium and that there appears to be no tendency to form continuous grain boundary carbides. This very important processing consideration may more than compensate for the low core K_{Ic} (about 40 MPa$\sqrt{m}$) and high alloy content.

CBS1000M

This alloy is based on a composition developed by Timken as a carburizing grade bearing steel (CBS1000). Unlike CBS600, it is a secondary hardening carburizing grade steel with a temperature capability comparable to Vasco X-2M. The commercial austenitizing temperature is about 1100C (same as for M50 NiL). The K_{Ic} for VAR material is about 40 MPa$\sqrt{m}$. This value is expected to be significantly higher for VIM-VAR material. The fracture toughness as well as the hardness is retained after exposure for 1000 hours at 315°C. Early consideration of this alloy was rejected because of its low Charpy energy at 40C but here again this low value is expected to increase for VIM-VAR material. A major incentive to use this steel as a carburizing grade gear steel are manifold and include considerable experience in its use as a carburized bearing steel, the stability of both hardness and fracture toughness after 1000 hours aging at 315C, and the ease of carburizing. The modification of this alloy by AMAX suggests that the composition of CBS1000M can be optimized with advantage. This work will be discussed below.

AMAX B

This alloy was developed at amax metals Research Laboratories in Ann Arbor, Michigan under contract to AMMRC. The program grew out of contacts at National Materials Advisory Board meetings which focussed on the shortcomings of several candidate steels for use as carburized gears for high temperature service in helicopters (Ref 3). The initial program concentrated on a comparison of carburized candidate high temperature steels (shown in Table I), and the investigation of six experimental steels. This program concluded that an experimental composition similar to CBS1000 had the highest impact fracture strength, even higher than AISI 9310. A follow-on program had as its major goal to optimize this

experimental composition by variations in the concentrations of silicon, molybdenum, and nickel. The objectives were to produce by composition and heat treatment, fracture characteristics similar to AISI 9310 and a minimum surface (carburized) hardness of 58 R_c both before and after 1000 hour exposure at 315C. AMAX B satisfies these criteria with a composition almost 2% less molybdenum and almost 1% less nickel than CBS1000. This program, now completed, should be complemented by tests of long time stability at 230C (as for X-53). Successful completion of these tests suggest that this alloy, with every indication of stability up to 315C, may well become the prime candidate for the replacement of Vasco X-2M. Considerable processing studies and gear tests (including rolling contact fatigue, single tooth bending, and 4 square gear testing), however, must be carried out before the final decision can be made on this alloy.

SUMMARY

The use of high temperature high performance gears in helicopters has wide acceptance within the helicopter industry in the USA. With the exception of CBS600, the efforts have for the most part focussed on low carbon modifications of secondary hardening tool steels. Vasco X-2M is the only high temperature gear steel currently flying in the main drive train of helicopters in the USA. Because of the extensive engineering data base associated with VASCO X-2M, the long history of manufacturing development, and the significant improvements in metallurgical parameters over the past 15 years, Vasco X-2M would be the obvious choice for a high temperature gear steel requirement for critical aircraft applications. This does not mean that other steels should not be considered. In fact, the two additional major candidate steels were chosen principally because they had a higher fracture toughness than Vasco X-2M. In the past several years considerable metallurgical, engineering, and manufacturing information has been developed for CBS600 and X-53 by Sikorsky and Bell respectively.

CBS600 is a carburizing bearing steel developed by Timken and has had considerable use in roller bearings for applications up to 230C. It is the only candidate high temperature gear steel which is not a secondary hardening tool steel. Except in special applications, it is not expected to find wide application in helicopter drive trains because (1) its marginal temperature capability and (2) its high fracture toughness is being approached by secondary hardening tool steels. For example, the fracture toughness of Vasco X-2M was gradually increased from about 45 MPa$\sqrt{m}$ to 70-80 MPa$\sqrt{m}$ by improved melting practice (VAR, VAR-VAR, and VIM-VAR).

X-53, had initially a higher fracture toughness than Vasco X-2M in the vacuum arc remelted (VAR) condition. With the improvement in fracture toughness of both alloys with increased cleanliness (VIM-VAR), the difference in fracture toughness is less but still significant. A temper embrittlement transformation was first identified in X-53. This embrittlement appears to be dependent on the austenitizing temperature and the lowest value of toughness (as determined by room temperature Charpy energy values) with increased time or temperature of aging is higher for the VIMVAR than the VAR material. Long aging times (2000 hours) at 230C result in improved toughness and less scatter in the data. Experiments are currently underway at AMMRC to determine the practicality of duplicating the toughness values for specimens aged for 2000 hrs. at 230C with those obtained on specimens aged for much shorter times at 315C. It appears that X-53 could become a competitor to Vasco X-2M as the next aircraft quality high temperature gear steels but it is not clear if the possible benefits are worth the added costs to carry out the necessary engineering and manufacturing studies.

M50NiL is being developed as a high fracture toughness bearing steel. As a gear steel, the 40 MPa$\sqrt{m}$ fracture toughness, makes it a poor last of the materials being considered. M50NiL, however, can be readily vacuum carburized and in the integral gear/bearing configurations being currently designed into advanced transmission systems may well be considered as a high temperature gear material especially in applications where operating temperatures in excess of 300C are expected. It should be reiterated that the value of fracture toughness necessary for gear steels is not known.

CBS1000M was also developed by Timken as a higher temperature carburizing bearing steel than CBS600. It has comparable fracture toughness to M50NiL and can be carburized as readily as AISI 9310. At this time, it would not appear to be a strong candidate for use as an aircraft quality high temperature gear steel despite its ability to retain both its hardness and toughness after 1000 hours aging at 315C. Although it has been widely used as a high temperature carburizing grade bearing steel by Timken, considerable engineering and manufacturing effort must be expended before it can be seriously considered for use as a gear steel. The hoped for payoff vis a vis Vasco X-2M does not appear to be sufficiently great to justify the added costs.

AMAX B was developed as a compositional modification of CBS1000 at the AMAX Metal Research Center in Ann Arbor, Michigan. The major requirements for the development of this alloy were the retention of both the carburized surface hardness and fracture toughness after an aging cycle of 1000 hrs. at 315C. The approximate 2w/o less molybdenum and 1w/o less nickel in AMAX B would suggest a higher fracture toughness value than for CBS1000. Although no ASTME399 valid K_{Ic} test has been performed, the K_{Ic} for AMAX B appears to be considerably higher than for CBS1000. A great deal of metallurgical, engineering, and manufacturing effort must be carried out on AMAX B before it can be even considered for an aircraft quality high temperature gear application. It is felt, however, that its ease of carburizing, low alloy content, and high fracture toughness in addition to retention of both hardness and toughness after exposure for

1000 hrs. at 315C make it a very good candidate to replace Vasco X-2M. In fact, it may well be an excellent candidate steel for the integral gear/bearing/spline configurations found in advanced drive systems and be worth the cost of development. Before a more definitive projection of this alloy can be made, however, metallurgical studies (relatively low cost) must be carried out. The use of high temperature high performance carburized gears in advanced helicopter drive systems is expected to dominate the next decade. With the addition of elevated temperature transmissions, this domination is expected to accelerate. While Vasco X-2M, for technical as well as economic reasons, is the logical selection (at this time) for critical aircraft high temperature gear applications, it is expected that, within the next decade, at least one of the candidate gear alloys will compete successfully with Vasco X-2M. These successful candidate steels will no doubt be secondary hardening tool steels because of their greater scuffing resistance than AISI 9310 and their improved fracture toughness vis a vis VASCO X-2M.

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TABLE 1 - CHEMICAL COMPOSITIONS OF GEAR STEELS

Steel	C	Si	Cr	Ni	Mo	W	V	Cu
AISI 9310	.1	.25	1.25	3.5	-	-	-	-
VASCO X-2M	.15	1.00	5.0	-	1.5	1.5	.5	-
CBS 600	.20	1.00	1.48	-	1.0	-	-	-
CARTECH X-53	.10	1.00	1.00	2.0	3.0	-	2.0	2.0
CBS 1000	.15	.5	1.00	3.0	4.0	-	.5	-
AMAX B	.12	1.00	1.00	2.25	2.25	-	0.5	-
M50NiL	.12	.25	4.5	3.0	4.0	-	1.25	-

TABLE 2 - SURFACE HARDNESS OF CARBURIZED STEELS BEFORE AND AFTER EXPOSURE FOR 1000 HOURS AT 315C. (REF. #4)

Surface Hardness of Carburized and Hardened Steels Hardness, R_C

Steel	Before[a]	After[a]
AISI 9310	61.2	52.3
CBS 600	65.4	58.8
VASCO X-2M	58.7	61.0
CARTECH X-53	58.4	59.5
CBS 1000	58.6	58.4
AMAX B	60.0	60.0
M50NiL	61.0	61.0

a - before and after exposure for 1000 hours at 315C.

TABLE 3 - FRACTURE TOUGHNESS OF CARBURIZED STEELS BEFORE AND AFTER EXPOSURE FOR 1000 HOURS AT 315C (REF. #4)

Fracture Toughness in Carburized Cases (Corrected for Residual Stress Effects)

Steel	Carbon Content, %	Fracture Toughness, K_{I_c}[a] MPa$\sqrt{m}$ (ksi$\sqrt{in.}$) Before Exposure [b]	After Exposure	Change,[c] %
CBS600	0.50	53 (48)	47 (43)[d]	(10)
	0.75	45 (41)	36 (33)[d]	(8)
CBS1000	0.50	44 (40)	33 (30)	(10)
	0.75	21 (19)	26 (24)	(25)
X2(M)	0.50	45 (41)	21 (19)	(53)
	0.75	25 (23)	13 (12)	(48)
X-53	0.50	48 (44)	22 (20)	(54)
	0.75	36 (33)	23 (21)	(36)
SAE 9310	0.50	42 (38)	Too Soft	--
	0.75	27 (25)	Too Soft	--

[a] K_{I_c} determined using specimens with short crack lengths.

[b] Exposure to 315 C (600 F) for 1000 hours.

[c] Parentheses indicate the change was a decrease (negative).

[d] Some softening occurred but remained above HRC 58 (see Table 2).

TABLE 4 - DIFFUSION CALCULATIONS

$$D = D_o \exp(-Q/RT)$$

$$D_1 t_1 = D_2 t_2$$

$$t_2 = \exp(Q/RT_2 - Q/RT_1)\, t_1$$

D = DIFFUSION CONSTANT

Q = ACTIVATION ENERGY

R = GAS CONSTANT

T_1 = 230C + 273 = 505K

T_2 = 315C + 273 = 588K

t_1 = 2000 hrs. (at 230C)

t_2 = X hrs. (at 315C)

FOR Q = 57,700 Cal.; t_2 = 0.67 hrs.

Q = 30,000 Cal.; t_2 = 30 hrs.

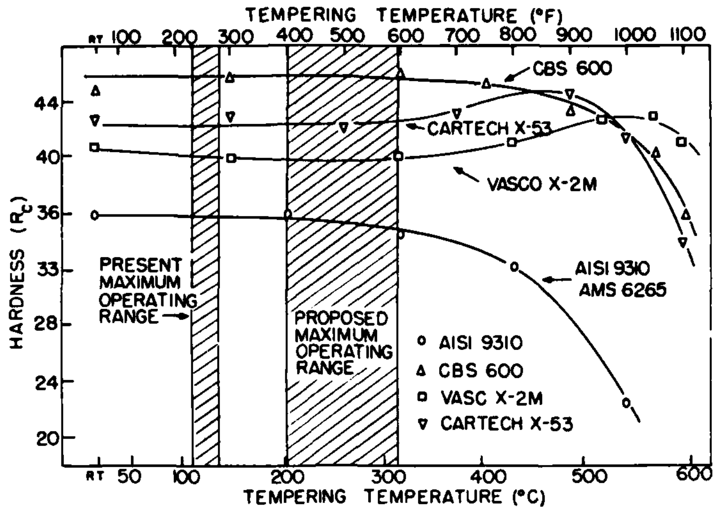

FIGURE 1 - EFFECT OF TEMPERING TEMPERATURE (2 + 2 HOURS) ON HARDNESS FOR SEVERAL GEAR STEELS

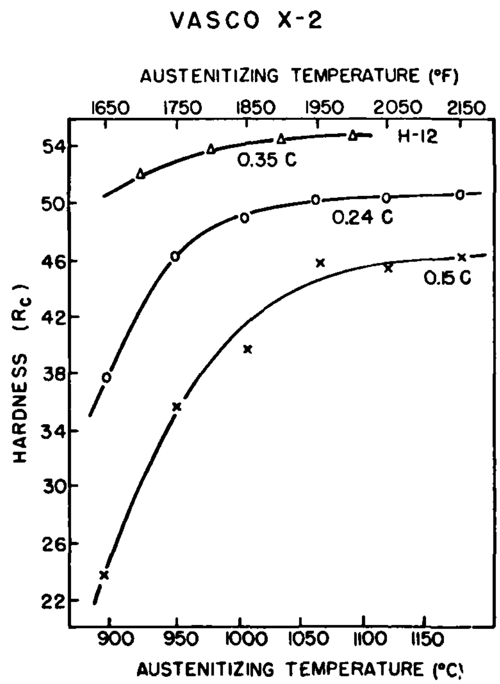

FIGURE 2 - EFFECT OF AUSTENITIZING TEMPERATURE ON HARDNESS FOR VASCO X-2

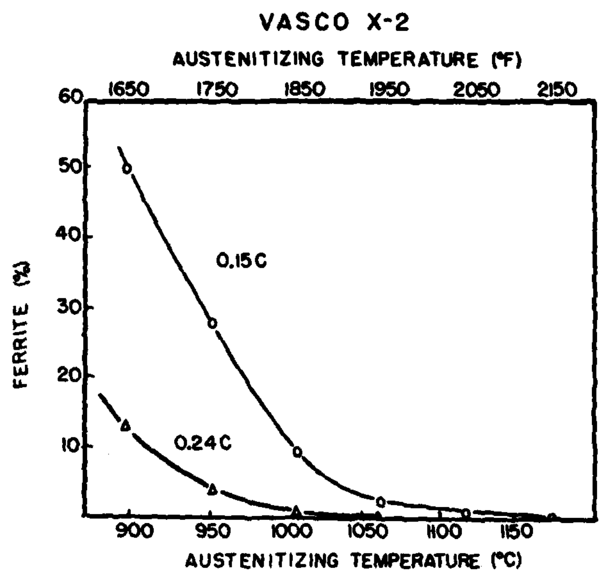

FIGURE 3 - EFFECT OF AUSTENITIZING TEMPERATURE ON AMOUNT OF PRIMARY FERRITE FOR VASCO X-2

900C 500X

950C 500X

1000C 500X

1050C 500X

FIGURE 4 - EFFECT OF AUSTENITIZING TEMPERATURE ON THE MICROSTRUCTURE OF VASCO X-2M

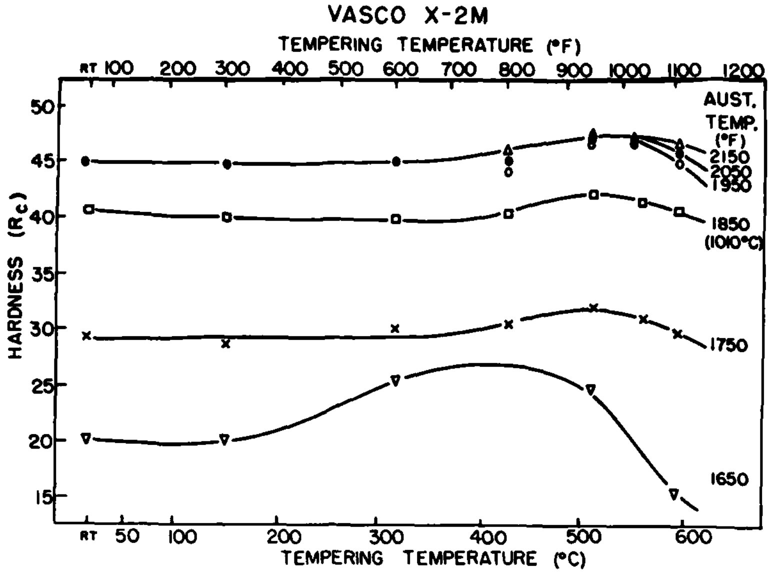

FIGURE 5 - EFFECT OF TEMPERING TEMPERATURE ON HARDNESS OF VASCO X-2M FOR SEVERAL AUSTENITIZING TEMPERATURES

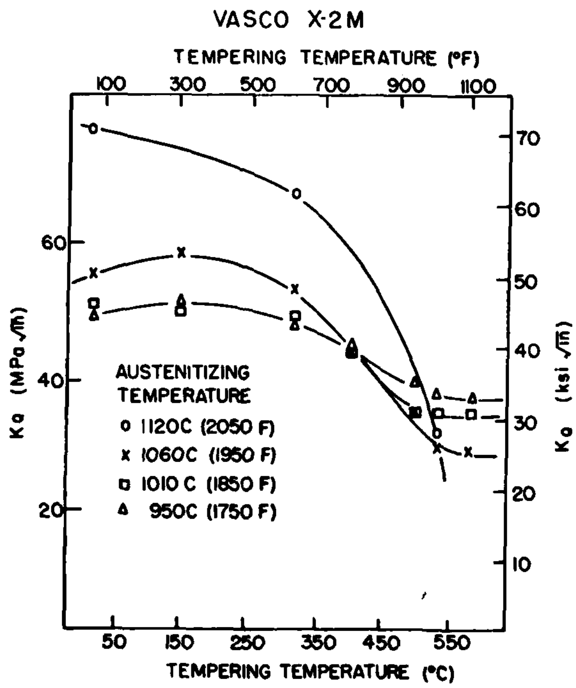

FIGURE 6 - EFFECT OF TEMPERING TEMPERATURE ON K_Q FRACTURE TOUGHNESS OF VASCO X-2M

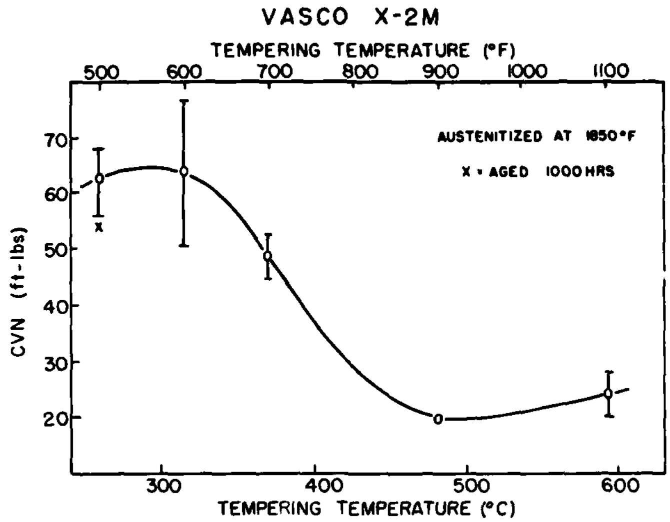

FIGURE 7 - EFFECT OF TEMPERING TEMPERATURE ON THE CHARPY ENERGY OF VASCO X-2M

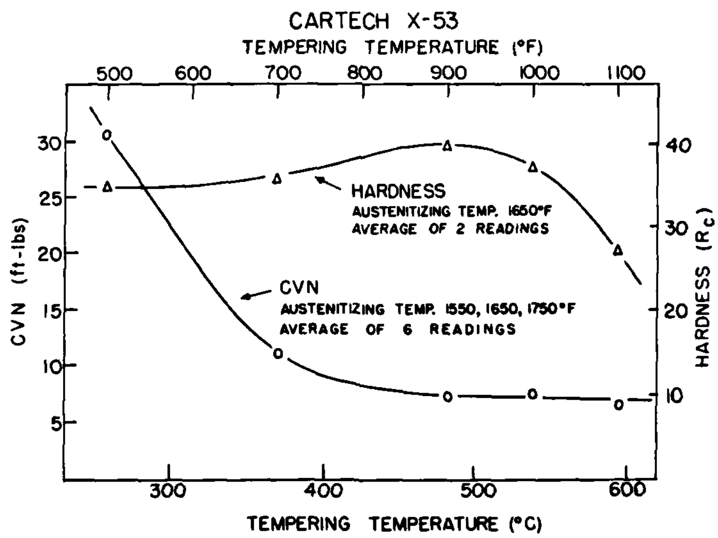

FIGURE 8 - EFFECT OF TEMPERING TEMPERATURE ON THE HARDNESS AND CHARPY ENERGY OF X-53

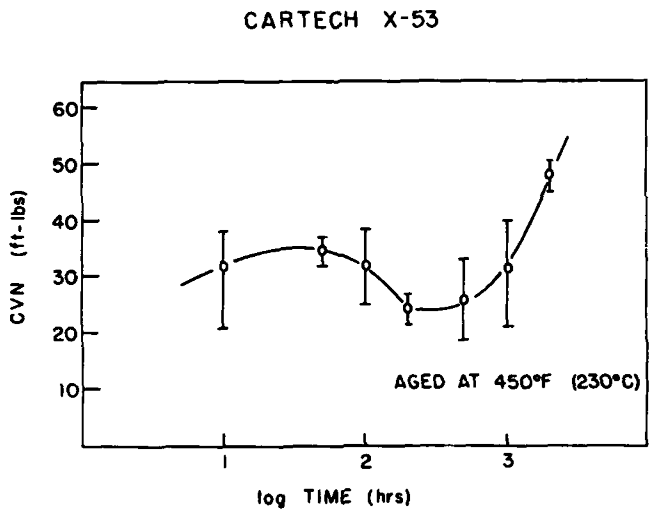

FIGURE 9 - EFFECT OF AGING TIME AT 230C ON CHARPY ENERGY OF X-53

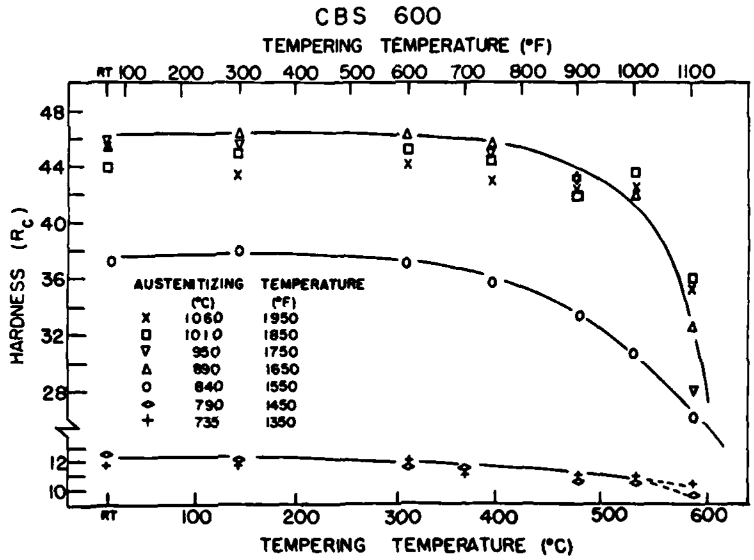

FIGURE 10 - EFFECT OF TEMPERING TEMPERATURE ON HARDNESS OF CBS600 FOR SEVERAL AUSTENITIZING TEMPERATURES

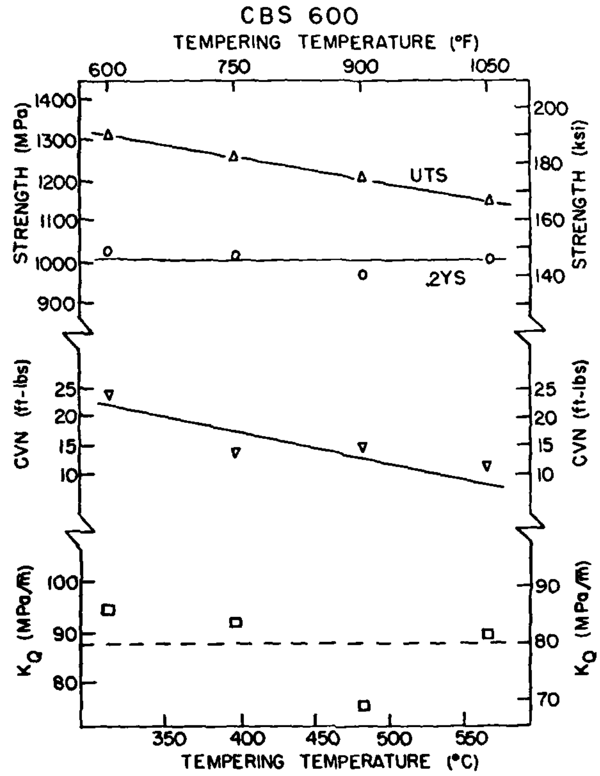

FIGURE 11 - EFFECT OF TEMPERING TEMPERATURE OF STRENGTH AND TOUGHNESS OF CBS600

AD-P005 062

THE ROLE OF RESIDUAL STRESS IN THE PERFORMANCE OF GEARS AND BEARINGS

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SUMMARY

Residual stresses are an inevitable consequence of the manufacture and service conditions to which mechanical components are subjected. In this paper, a wide range of evidence is presented to show the decisive effect of residual stress, both pre-existing and service induced, on the performance of gears and rolling element bearings.

The results of measurement of residual stresses arising from a range of manufacturing procedures are presented, particular emphasis being placed on carburized steels. The effect of such stresses on fatigue performance is demonstrated. Possible causes of residual stress change during service are reviewed and the results of new experimental and theoretical work on the role of residual contact stress in a number of relevant tribological failure modes are presented.

INTRODUCTION

Interest in the topic of residual stress comes in waves. Such waves can be created by a wide variety of circumstances. Sometimes the originating disturbance is a practical problem such as was created by stress corrosion following the introduction of high strength aluminium alloys or by the discovery of the effects of grinding abuse in hardened steels. On the other hand, waves of equal ferocity have been generated by the development of new investigative techniques such as the new "fast" Xray diffraction methods and equally as often by theoretical advances such as the application of shakedown theory to rolling contact in the early 1960's.

It is remarkable, however, how little constructive interference there has been between these various sources of interest. In this paper, an attempt is made to review the role of residual stress in performance of gears and rolling element bearings. Particular emphasis is given to relating experimental and theoretical determination of residual stresses to the outcome in terms of performance. To this end, the paper is divided into two sections. The first deals with the - perhaps more widely accepted and understood - topic of the effect of pre-existing residual stress on performance. In the second part of the paper consideration is given to residual stresses arising during service. A new approach to residual stresses in plastically deformed asperities is presented and its consequence on tribological failure modes in aircraft components is discussed.

It is hoped that any ripples of interest which may thus be generated will not be too swiftly attenuated, whatever their wavelength!

PRE-EXISTING RESIDUAL STRESS

When a component has been manufactured, it practically always contains a locked-in stress distribution. In this section the nature of this pre-existing residual stress, its measurement and its effect on performance are considered.

Permitted Stress States

A residual stress state may be defined as one in which the boundary loads on the body in question are zero. Residual stress states are elastic, that is to say that the yield criterion is not exceeded by the residual stresses, and they obey the law of equilibrium. It is instructive to consider some of the restrictions this places on possible residual stress states. In cartesian coordinates, the equilibrium law is (in the absence of body forces) (1):

$$\begin{aligned}
\frac{\partial \sigma_x}{\partial x} + \frac{\partial \tau_{xy}}{\partial y} + \frac{\partial \tau_{xz}}{\partial z} &= 0 \\
\frac{\partial \tau_{yx}}{\partial x} + \frac{\partial \sigma_y}{\partial y} + \frac{\partial \tau_{yz}}{\partial z} &= 0 \qquad (1) \\
\frac{\partial \tau_{zx}}{\partial x} + \frac{\partial \tau_{zy}}{\partial y} + \frac{\partial \sigma_z}{\partial z} &= 0
\end{aligned}$$

If we consider a uniform residual stress distribution near the surface of an infinite half space - a good approximation if the residual stress has arisen from a homogeneous surface treatment of a thick, flat, component - then the derivatives with respect to x and y will disappear giving:

$$\frac{\partial \tau_{xz}}{\partial z} = \frac{\partial \tau_{yz}}{\partial y} = \frac{\partial \sigma_z}{\partial z} = 0 \qquad (2)$$

where z is in the direction of the normal to the free surface.

As all these stresses must be zero at the surface because it is unloaded, then they are identically zero throughout and the only stresses which can exist are σ_x, σ_y and τ_{xy}. The body is in a state of plane stress.

A similar argument can be made for a uniform cylindrical body by expressing the equilibrium law in cylindrical coordinates. For a uniform, cylindrically symetric, stress state we have:

$$\frac{d\sigma_r}{dr} + \frac{\sigma_r - \sigma_\theta}{r} = 0 \qquad (3)$$

This means that the stress perpendicular to the surface, σ_r, is not zero except at the surface but satisfies equation (3). If the surface of the body is at $r=R_o$, then since $\sigma_r=0$ at $r=R_o$ then the sign of σ_r just below the surface depends on the sign of σ_θ (Figure 1).

If σ_θ is compressive, for example, then σ_r will be tensile just below the surface of a cylinder ($r<R_o$) but compressive below the inner surface of a tube ($r>R_o$). Usually, the magnitude of σ_r is small in practice, but an important exception to this arises when a cylinder of small diameter is case hardened (leading to compressive, i.e. negative σ_θ). The magnitude of the tensile σ_r, component can then be quite large, and will rise to a maximum at the case-core boundary. Some case-core separation problems are probably related to this residual stress.

It may be felt by the reader that the necessity for residual stress distributions to satisfy equilibrium is something of a truism. However, many published experimental residual stress distributions do <u>not</u> appear to satisfy this law. For example, Meda et al (2) report residual stress measurements below the surface of a cylindrical body for which $\tau_{r\theta}\neq 0$. If this measurement were correct it would imply that the stresses were not cylindrically symmetric and hence should vary along the cylinder; such variation was not reported however. The reasons for this type of discrepancy probably lie in the measurement techniques. These are discussed briefly in the next section.

Measuring Residual Stresses

It is not always appreciated just how many different, but related quantities are covered by the description "residual stress". A large number of techniques exist for measuring residual stress and of these only one measures the fundamental quantity familiar to engineers. This technique involves measurement of strain relaxation during controlled, incremental removal of material. The commonest variant of the technique is the hole-drilling method, described by Bathgate (3) in which a hole is formed progressively in the surface and the radial strain relaxation measured using a strain gauge rosette. The technique can be made quite reproducible with care but suffers from the disadvantage of poor resolution of stress gradients and of very low sensitivity for depths greater than the hole diameter. It is also, of course, destructive though is not regarded so by some heavy industries where small holes can be tolerated.

Xray diffraction (XRD) techniques are also widely used for residual stress measurement and have become more popular in recent years with the devleopment of more rapid, automated equipment. However, XRD does not measure the same quantity as the destructive techniques and in many circumstances gives results which differ, sometimes by a large margin. The principal of the Xray technique is will understood and is shown in diagrammatic form in Figure 2. A recent review of theoretical aspects by Dölle is highly recommended (4). Measurements of normal displacement of crystal interpanar spacing are made as a function of direction. These may then be converted into stresses using a knowledge of the local elastic properties which must be obtained from a separate calibration experiment.

The XRD method has a number of attractive attributes. One is that it can resolve high stress gradients which can be of great significance in surface treatment technology and it can also detect residual shear stresses within the penetration of the Xray beam. The principal of this is shown in Figure 3. The presence of the shear stress component gives rise to different interplanar spacings with respect to forward or backward specimen rotation. However, Xrays are diffracted only by crystalline material of a particular phase which may not be in the same state of stress as non-crystalline regions (such as subgrain boundaries) or as material of other phases. When and whether such effects are important appears to depend strongly on the material and its strain history. A review of these effects which have been dubbed "pseudomacrostress" has been given by Cullity (5), who shows that magnetic effects, which are also sensitive to the stress, behave as would be expected from the XRD stress measurement.

Residual Stresses and Fatigue in Carburised Steels

In this section, the results of study of the fatigue properties of carburised steels is presented in conjunction with extensive investigation of the role of residual stress. The importance of a complete stress analysis, which includes consideration of residual stresses is demonstrated.

The purpose of the work was to examine the high cycle fatigue behaviour of gear material under conditions as close as possible to those encountered in helicopter gears. In particular, the related variables of tooth-root stress concentration, of carburised case depth and of applied mean stress, were arranged in such a way as to provide a realistic distriubtion of applied stress whilst still enabling the use of a simple, axially loaded, fatigue specimen. Of particular interest were the tooth root stress characteristics of the Wildhaber-Novikov conformal gears, which are used in the main gearbox of Westland Lynx and Westland 30 helicopters. Details of the tooth root stresses have recently been published by Astridge et al (6) and feature applied mean stresses in the compressive region.

The specimen is shown in Figure 4. Results of a 2-D finite element analysis of the specimen is shown in Figure 5. The stress concentration associated with the notch has a maximum value of about 1.7. Note that the region in which the applied stresses exceed the average stress in the reference section is confined to the carburised case. To find the actual stresses in this region we therefore require a knowledge of the residual stresses in the case.

Manufacture of the test specimens was carried out by techniques closely following those used for real components. The specimen notch was manufactured in the same manner as a preformed gear tooth root; that is by machining followed by heat treatment (case hardening) and finally shot peening. The details are shown in Table 1. The heat treatment adopted is also shown in Table 1. The effect of subzero treatment was investigated by omitting this process on half the specimens.

Residual stresses were measured using an Xray diffraction technique. By selection of suitable diffraction peaks it was possible to obtain residual stress values for both the metallurgical phases (martensite and austenite) present in the specimen case.

The specimens were tested under tensile, zero and compressive applied mean stresses, the ratio of alternating to mean load being held constant througout each series. The testing frequency was approximately 150 Hz. The results are shown in Figure 6 in the form of a Goodman diagram. Here the nominal endurance applied stress range (ignoring stress concentration) is plotted against the nominal mean stress (ignoring residual stress). The mean endurance limits shown were calculated, using a standard curve shape, from the individual fatigue lives.

During the testing it became evident that two types of failure were occurring. One of these involved fatigue initiation in the notch, close to the surface, usually at a depth just below the shot peened layer. The other form of failure originated in the uncarburized core of the specimen, at a number of locations. The proportion of failures obtained of each type was found to depend on the applied mean stress, there being more core-originated failures at compressive applied mean stress.

The performance of the 4% NiCrMo steel is superior under all conditions tested to the 3½% NiCrMo, the preferred steel in the U.S. Subzero treatment had little effect.

The results of the Xray diffraction work are shown in Figure 7. The upper part of the figures show the proportion of retained austenite present as a function of depth. The proportion of this phase is reduced but not eliminated by the subzero treatment.

A complex residual stress state is present. Very high compression is present at the surface and persists to a depth of about 0.1mm. This is the area affected by shot peening. At greater depths but still within the carburised case, a more moderate compression is present in the martensitic phase, but tensile stresses are present in the austenite. The stress in the austenite could not be measured for depths below 0.3mm for the subzero treated specimens and about 0.65mm for the untreated specimens because the diffraction peak became too weak, with declining austenite content, to locate sufficiently precisely. Subzero treatment, although reducing the total amount of austenite present, also has the effect of increasing the tensile stress in this phase. On the other hand, the compression in the martensite is increased by subzero treatment. In the core, the stresses are tensile.

The combined effect of the notch and the residual stresses are that both alternating and mean stresses differ between the two failure origin locations. In Figure 8 the real stresses at the endurance limit are plotted in the form of a Smith diagram for the standard material condition. Two series of approximately straight lines are obtained which coincidentally converge to the proof stress value for the core. Portrayal of the data in this form provides all the fatigue information required whilst at the same time allowing extrapolation to cases where the residual stress state is not the same. An important example of this occurs if the proportion of case to core varies from that used in the present experiments. Higher proportions of case give rise to higher tensile stresses in the core.

Residual Stress and Critical Defect Size

All materials contain defects. The size and distribution of such defects have a very substantial effect on fatigue performance especially for high strength steels of the type used for aircraft tribological components. In this Section an example is given of the analysis of the fatigue behaviour of a gear containing such defects in order to demonstrate the large effect of residual stress.

A service failure had occurred of a pinion gear. Investigation showed that the origin of the failure was in the (uncarburised) bore, a region which was known to be very mildly stressed. However, the initiation of the failure was associated with a small pre-existing crack-like defect which had probably arisen during manufacture. Defects of this nature could be shown to reduce fatigue life in coupon tests but it was required to know whether such a defect could propagate under service conditions. An analysis was therefore undertaken, using linear elastic fracture mechanics, in order to determine the effect of service stresses on such defects.

It soon emerged that one of the major unknowns was the residual stress. A tensile residual stress acting transversely to the defect would allow crack opening over a much larger proportion of the stress cycle and would thus accelerate propagation. Equally, tensile stress would allow smaller defects to

propagate at a stress which might otherwise be below the threshold. The effect of a constant tensile stress on the critical defect size to give the failure life is shown in Figure 9. Measurement of the actual residual stress in the bore of the gear shaft proved impossible but test pieces of similar section treated in the same way showed substantial tensile stresses of approximately 300 MPa. The critical flaw size was therefore of the order 10^{-1} mm, comparable with that of the observed defects.

The effect of flaw size on life for different constant residual stresses is shown in Figure 10. The residual stress has an overwhelming effect on performance. This investigation culminated in the removal both of the damaging residual stress and of the defects, by modification of the manufacturing route. At the same time, a new differential eddy-current inspection technique was introduced to give further assurance of freedom from surface flaws.

Residual Stress and Rolling Contacts

The effect of residual stress on concentrated contacts is a more difficult problem than that considered in the last section because of the complexity of the applied stress field. The simplest form of the problem is the effect on the static strength of concentrated contact. This merely requires the superposition of residual and applied stress fields and the application of a yield criterion to the resultant. Hills and Ashelby (7) and Broszeit et al (8) have recently persued this line of work. In general, uniform compressive residual stress is beneficial, despite the compressive nature of the applied stresses, because it has the effect of reducing the difference between the principal stresses in the region beneath the contact and hence reducing the maximum shear stress. Yield is consequently inhibited. The compressive residual stresses produced by carburising, nitriding, mild grinding, shot peening etc therefore act to increase the static load carying capacity of surfaces.

However, the performance liimiting factor for many aircraft gears and rolling element bearings is not static behaviour but pitting fatigue. This phenomenon is still not well understood despite having been the subject of much research. It does seem, however that compressive residual stress can improve pitting life (9). Equally, tensile stresses can reduce performance although it seems that the effect varies withthe direction of the tensile stress. Czyzewski (10) investigated the effect of a tensile hoop stress in a bearing race such as may occur when an inner race is shrink fitted onto a shaft or when an outer race is subjected to high centrifugal forces. He found a large reduction in life together with a change in cracking mode to give fracture of the race rather than pitting. Foord et al (11) and more recently Dousinas (12) have applied tensile stress perpendicular to the rolling direction in combined bending/rolling experiments with soft, high carbon steels. The results show a small life reduction.

Much work still needs to be done in this area both in relation to residual and to combined applied stresses. One problem is that in pure rolling, failure does not occur until applied loads approach the elastic limit. This means that the real stress field changes during running. Even when sliding is applied, some plastic deformation is still likely under conditions which enable surface asperities to come into contact. These possibilities are further explored in the next section.

SERVICE INDUCED RESIDUAL STRESS

A number of ways exist in which the residual stress state in a component can change during its service life. All can play a decisive role in gear and bearing failure modes.

Thermal Stress Relief

Carburised steels of the type used in many gear and bearing applications at low temperature (Westland practice for carburised 4% NiCrMo steel is to temper at 140°C). Some bearing rolling elements are tempered at temperatures as low as 125°C. In the case of carburised steels, the effect of heating the component at a higher temperature than this is two-fold. One effect is to change the hardness: the effect of overtempering on the microhardness profile of a carburised case is shwon in Figure 11. The surface hardness is in fact for moderate tempering periods at up to 200°C in this steel.

In addition, the residual stress distriubtion changes during overtempering. Kirk (14) showed that the beneficial compressive residual stresses produced during case hardening in the surface of the workpiece are rapidly relieved by thermal treatment in the range 100-200°C for ½% NiCrMo (8620 H) steel. He showed a corresponding reduction in fatigue properties. A similar investigation has recently been carried out at Westland for the 4% NiCrMo carburised steel, using 14 mm diameter specimens in rotating bending (zero applied mean stress). Again a significant reduction in fatigue performance has been observed even under circumstances where the surface hardness has not been reduced below the normally accepted minimum (Figure 12). Both these studies lead to the conclusion that satisfactory surface hardness does not imply that the performance capability of a component is unaffected after an overheating event; residual stresses may have been changed in a detrimental manner.

Residual Contact Stress

Of course it should not normally be the case that the operating temperature of a component exceeds its tempering temperature. However, when these temperatures are quite close as they often are for gear and bearing steels, changes which mimic overtempering may occur during running. The dark area sometimes observed in rolling element bearings of 1% CCr steel after long running times probably arise from this source. Thermal and/or cyclic softening allows plastic deformation to occur during prolonged running under the influence of applied loads.

The nature of the stress distribution which is induced by contact at loads above the effective yield has recieved much study and a review of the relevant theory has recently been presented by K.L. Johnson (14). If the applied loads exceed yield by only a small margin as is common in practical

situations, the approach of Merwin (15) may be used in which the total strain is equated to the elastic strain. This is reasonable because the plastic deformation is contained by elastic material to a small sub-surface region. A practical consequence of this is that such plastic deformation is impossible to detect metallographically. Nevertheless the stresses generated may be large. Figure 13 shows the residual stress distribution in identical rollers measured in one instance without running and in another after running in a disc machine at an applied Hertzian stress of 2.8 GPa at a slide roll ratio of 0.026. An increase in the case compression has apparently occurred during running. The nominal applied loads are close to the elastic limit but slightly below it. Evidently under real running conditions some plastic deformation has indeed occurred leading to the increased compression. Thermal and cyclic effects may again be significant here.

Residual Asperity Stresses

One of to the most important problems now being tackled by tribologists is the modelling of rough surface contact. Significant advances have recently been made in dry (16) and in lubricated (17) rough surface contact for situations in which the contact is fully elastic. However, it is well established from both theoretical and experimental evidence that plastic deformation may occur on an asperity scale when rough surfaces come into contact (18). Such plastic deformation will inevitably produce residual stresses. A simple way to calculate such residual stresses has recently been devised in a collaborative project between the Mechanical Engineering departments of Cambridge University and Imperial College, London (19). It is based on the assumption that the asperity loads may be high enough to be in the fully plastic range. Such conditions are believed to occur during gear tooth and race/roller contacts when the surfaces are rough and when the lubricant film thickness is low, especially though not exclusively, during running-in. This problem, of stress analysis of fully plastic contacts,has recently received much attention, the most popular recent approach being that of the finite element method. Curiously enough, no complete stress distributions based on this technique have been published, however. Fortunately, a much simpler approach is possible using slip-line field theory which allows an analytical solution to be obtained. Slip-line field theory has been used previously for asperity plastic deformation problems, notably by Green (20) Johnson (21) and Challen and Oxley (22). The residual stresses are obtained by using the elastic solution, for the same pressure and tangential force distribution, as is obtained from the plastic analysis: the asperity is "elastically unloaded".

The results of these calculations are reproduced in Figure 14. They show some startling effects. The unloaded contact surface is left in a state of high residual tension. Some subsurface stresses are also tensile, but not in the region immediately beneath the contact where a high, predominantly hydrostatic residual compression is predicted. The subsurface, tensile, residual stresses have a characteristic inclination related to the direction of tangential force. There is a close parallel between the direction of this (calculated) residual tension and the direction in which cracks are observed to form during the early stages of rolling fatigue and related froms of failure. Such cracks form at a shallow angle to the surface (Figure 15) and depend on the direction of applied tangential force in a similar manner to the calculated residual stresses. No detailed understanding of the mechanism of formation of rolling fatigue and micropitting cracks is currently available, but it does seem likely that the presence of residual tension acting perpendicular to the embryonic crack would assist its development in the manner mentioned earlier and hence favour crack formation in the observed direction.

In conclusion, it seems a mistake to assume, as is frequently done, that the operating conditions of tribological components are entirely in the elastic range. A knowledge of the effects of service-induced plastic deformation, leading to characteristic residual stress distributions is expected to make a major contribution to the understanding of contact failure modes which currently limit the performance of gears and rolling element bearings.

CONCLUSIONS

The aerospace industries of the world - quite correctly - expend substantial effort in order to determine the applied stress regime to which components and materials are subjected. However it is becoming increasingly evident that a full understanding of the performance-limiting failure modes requires consideration of residual as well as applied stresses. This is particularly true of tribological failure modes where the applied stresses are predominantly compressive.

This paper has given some examples of the use of residual stress analysis, both theoretical and experimental in the development of such understanding. It is to be expected that many potential improvements in performance and reliability of aircraft transmission systems could result from the development and exploitation of residual stresses, especially in the design of new materials and processes. A prerequisite to such advances is the ability to predict residual stresses and to verify such predictions by accurate measurements.

ACKNOWLEDGEMENTS

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1. Composition

Element		C	Ni	Cr	Mo	Si	Mn	S	P
Weight %	4% NiCrMo	0.15	4.15	1.12	0.25	0.26	0.37	0.005	0.006
	3½% NiCrMo	0.16	3.29	0.99	0.24	0.22	0.46	0.006	0.005

2. Manufacture

Both steels were manufactured by consumable electrode vacuum arc remelting.

3. Treatment

The specimens were carburised 925°C to give a surface carbon content of 0.8±0.05% to a nominal case depth of 1.5mm. The temperature was then reduced to 850°C for 1h before air cooling. After carburising the specimens were annealed at 650°C for 6h and furnace cooled. The remainder of the treatment was as follows:

Hardening:	Reheated to 790°C for 1h, oil quenched
Subzero:	Cool to -60°C for 1h
Tempering:	140°C 4h
Shot Peening:	Almen intensity 0.35mm A2 using S170 shot

4. Core Static Tensile Properties

	Ultimate tensile stress/MPa	0.2% Proof/MPa	Elongation	Reduction of Area
4% NiCrMo	1413	1312	15%	60%
3½% NiCrMo	1397	1253	14%	62%

Table 1 Material and Heat Treatment Details for 4% NiCrMo and 3½% NiCrMo Gear Steels

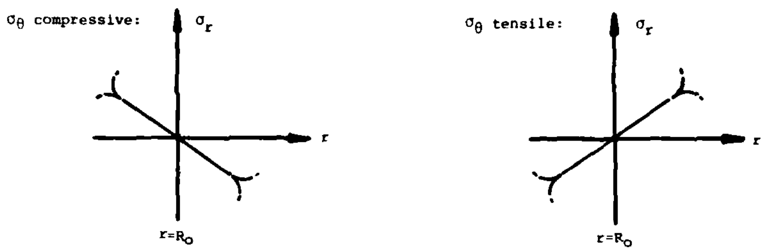
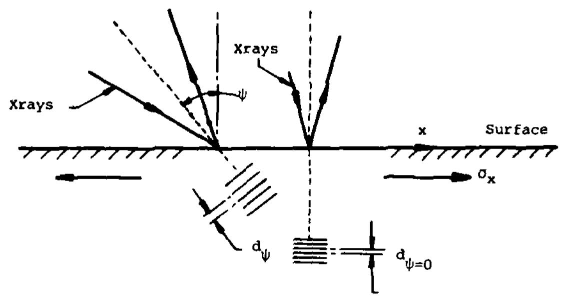


Figure 1
Radial Residual Stresses Near a Cylindrical Boundary



If σ_x is tensile $d_\psi > d_o$ $\quad \frac{d_\psi - d_o}{d_o} \propto \sigma_x$

Figure 2
Principal of Residual Stress Measurement by Xray Diffraction

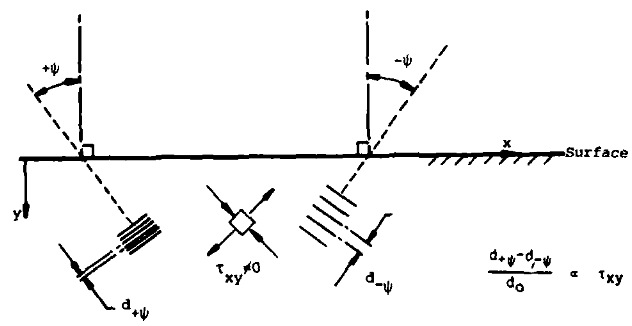


Figure 3
Principal of Residual Shear Stress Measurement by Xray Diffraction

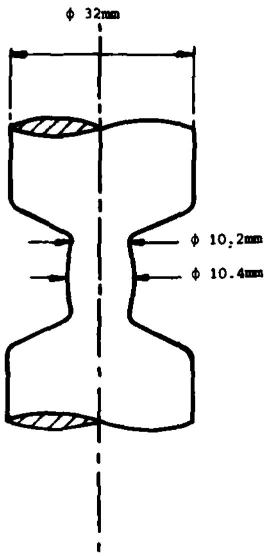


Figure 4
Direct Stress Specimen for Simulation of Tooth Root Fatigue Failure

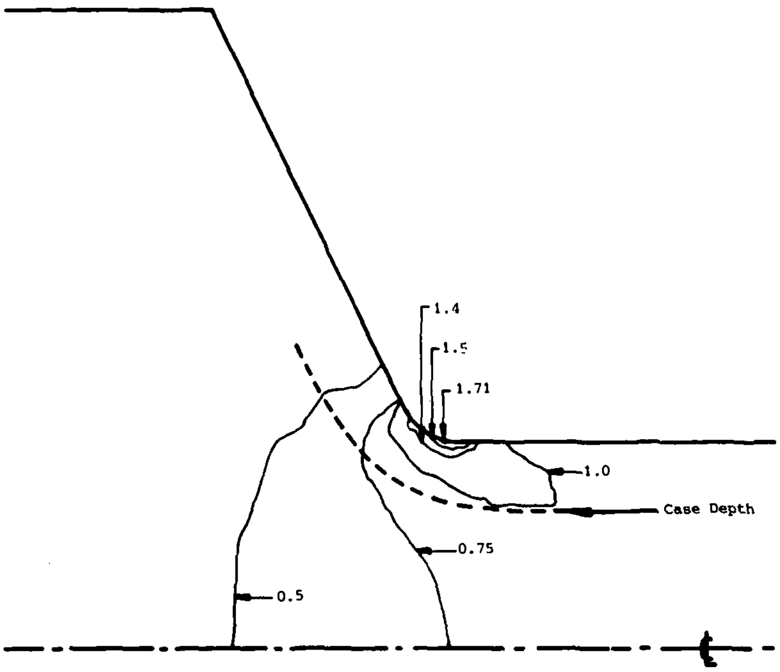


Figure 5
Finite Element Results for Fatigue Specimen Showing Contours of Stress Concentration

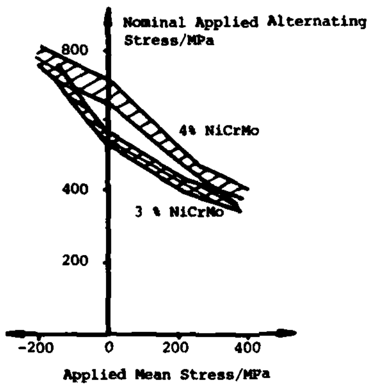


Figure 6
Goodman Diagram (RM Diagram) Showing Endurance Limits for 4% NiCrMo and 3½% NiCrMo Steel

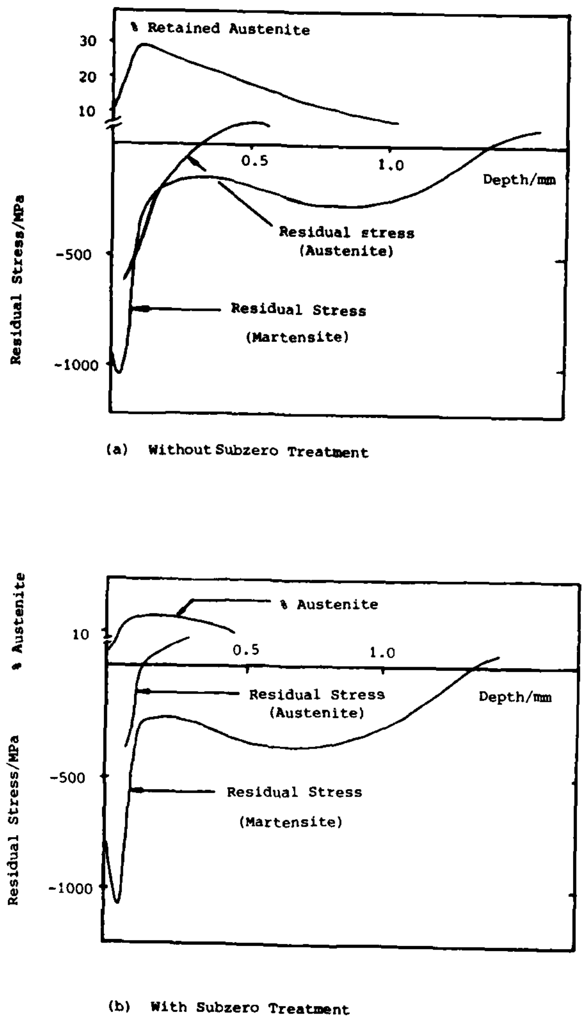


Figure 7
Residual Stresses and Retained Austenite Content in 4% NiCrMo Carburised Steel

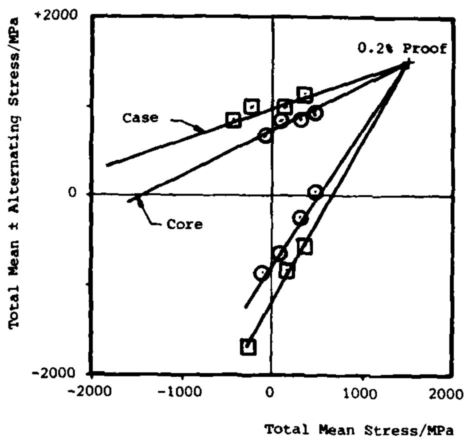


Figure 8
Total Stresses at Endurance Limit in 4% NiCrMo Steel

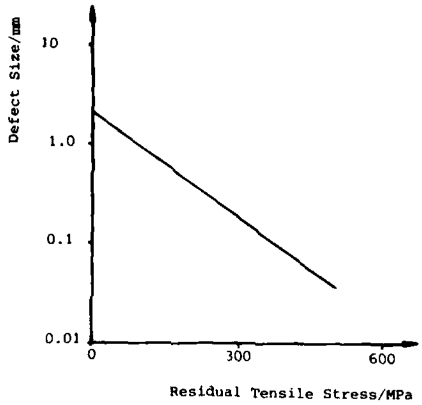


Figure 9
Effect of a Constant Tensile Stress on the Critical Defect Size to give a Constant Fatigue Life

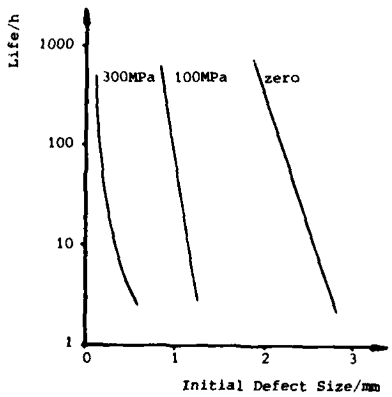
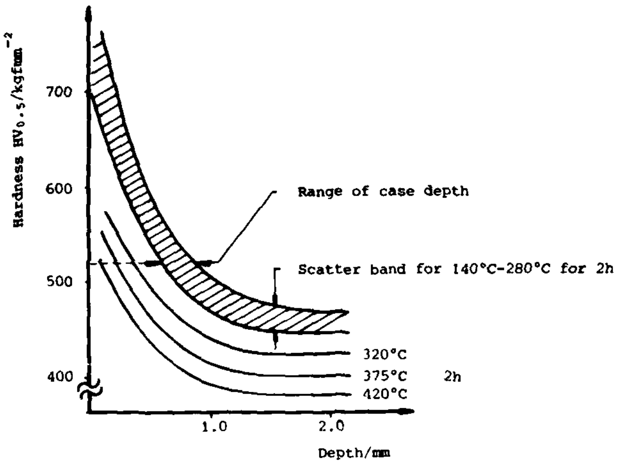
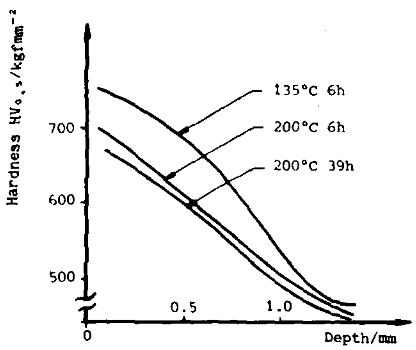


Figure 10
Effect of Defect Size on Fatigue Life for Different Constant Residual Stresses



(a) Short Term Tempering Data for 4% NiCrMo Steel
Carburised 8h, Reheated 790°C 30min, Oil Quenched,
Deep Frozen -65°C 1h, 140°C 2h, Retempered as shown



(b) Longer Term Tempering of 4% NiCrMo Steel, treated as for (a)

Figure 11
Effect of Tempering Temperature on the Case Microhardness Profile of Carburised 4% NiCrMo Steel

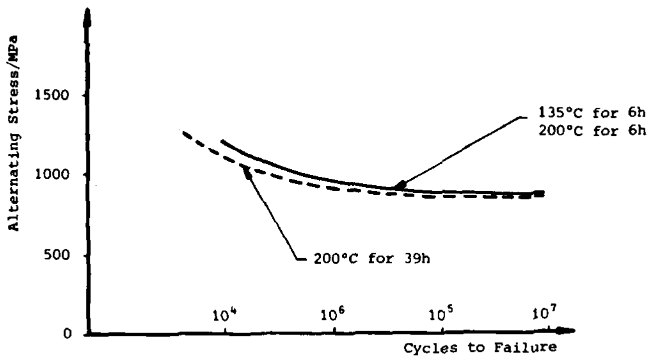


Figure 12
Effect of Overtempering on Rotating Bending Fatigue Properties of 4% NiCrMo Steel.
Test Section: 14 mm diameter

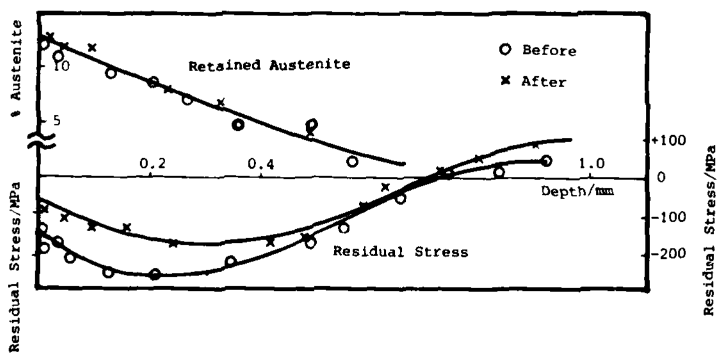
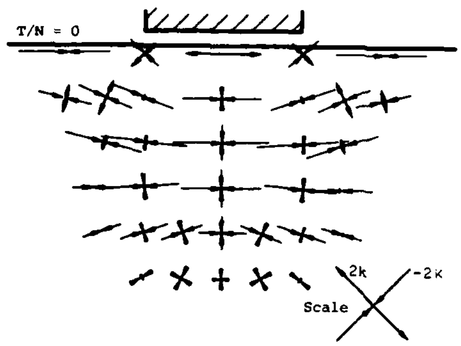
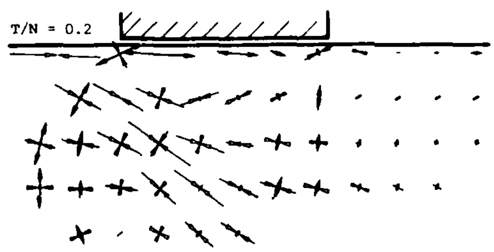


Figure 13
Residual Stresses in Disc Machine Roller Before and After Running at 2.8 GPa, 0.026 Slide/Roll for 2.2×10^6 Stress Cycles



(a) Zero Tangential Load



(b) Ratio of Tangential to Normal Load = 0.2

Figure 14
Residual Stress Distribution, Calculated Using Slip-line Field Theory, for a Plastically Deformed Asperity Contact
k = yield stress in shear

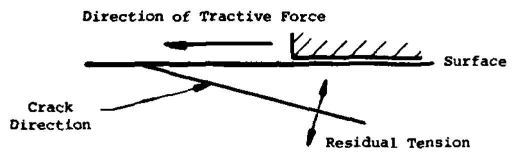


Figure 15
Crack Initiation Direction in Micropitting, Showing Relationship to Inclined Subsurface Residual Tension

THE LUBRICATION OF DYNAMICALLY LOADED CONCENTRATED HARD LINE CONTACTS: TEMPERATURE AND PRESSURE MEASUREMENTS

by

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SUMMARY

During more than 40 years, research into concentrated contacts with full film lubrication has been mainly restricted to stationary behaviour. It is not until recently, that dynamically loaded concentrated contacts have received attention from some investigators. With the advent of more sophisticated hardware and software, the interest from theorists into dynamically loaded concentrated contacts is growing. But, the literature of experimental results is even more scarce.
In this paper some results of temperature and pressure measurements in two types of dynamically loaded concentrated contact are reported, viz., a cam-flat follower contact and a cylindrical roller - outer race contact. The measurement technique utilizes miniature vapour deposited thin layer transducers and a counter current circuit and an amplifier. These first results are very encouraging. The transducers worked well, even under very adverse conditions of pure sliding and dynamic loads.

LIST OF SYMBOLS

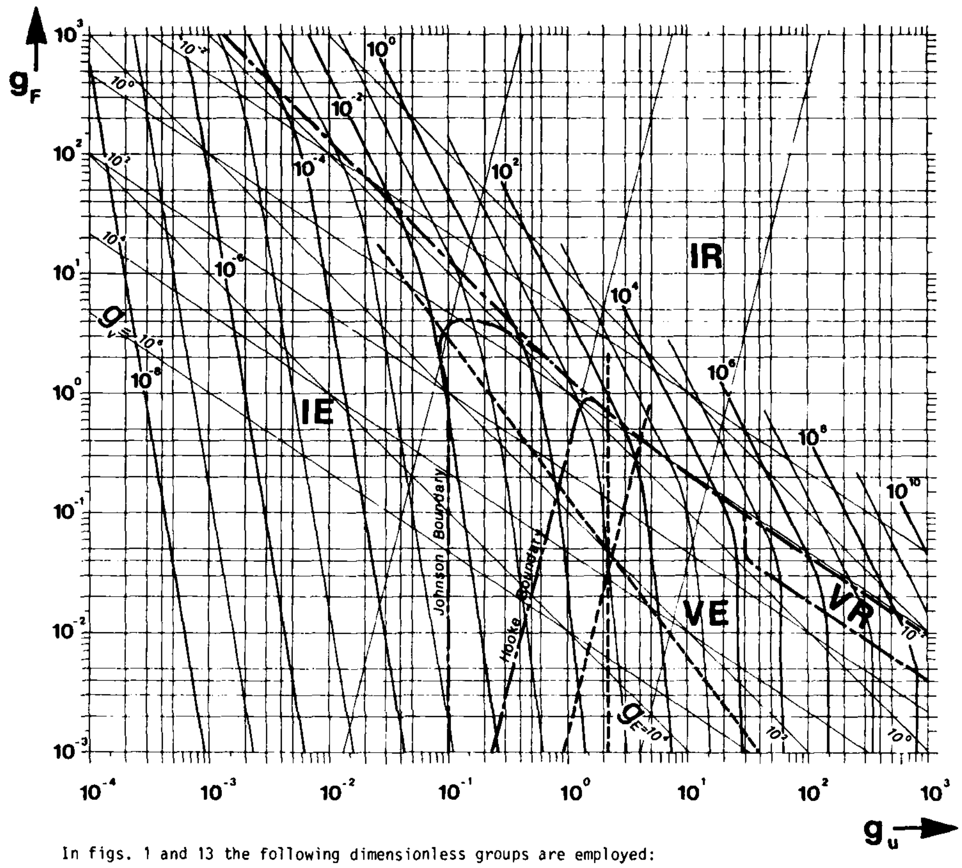
c	spring stiffness in cam-follower test rig	N/m
E_r	reduced modulus of elasticity	N/m²
F	load	N
g_u	speed number, dimensionless	—
g_F	load number, dimensionless	—
h_{min}	minimum film thickness	m
H	dimensionless film thickness	—
l	contact length	m
m	reciprocating mass in cam-follower test rig	kg
p	pressure	N/m²
R_r	reduced contact radius	m
t	time	s
T	temperature	°C
u	rolling speed	m/s
α	pressure viscosity coefficient	m²/N
Λ	relative film thickness	—
σ	RMS surface roughness value	m

1 INTRODUCTION

For some time, the lubrication of concentrated hard contacts has been considered as stationary and, mostly, as under fully-flooded, full-film conditions. Many examples can be found in the literature, e.g. Martin [1], Grubin [2][1], Blok [3], Dowson and Higginson [4], Dowson [5], Herrebrugh [6], Cheng [7] and, more recently, Hamrock and Dowson [8], [9] and Mohrenstein - Ertel [10][1]. In concentrated full film contacts, these presumptions are very often satisfied. Fig 1, which i taken from [11], shows a graphical summary of all these

[1] **Most researchers labor under a mistake in that they refer to Grubin as the original author of this paper. Recently, Lang and Oster have undoubtedly shown that Ertel [10] deserves full credit for this paper. Ertel wrote this famous paper in 1945.**

solutions for a line contact case. Therefore, the popularity of curve-fitting formulae like the Dowson - Higginson formula for line contacts, and Hamrock - Dowson formula for elliptical contacts, can be appreciated.



In figs. 1 and 13 the following dimensionless groups are employed:

$$g_u = \left(\frac{\alpha^4 E_r^3 \eta_0 u}{R_r}\right)^{\frac{1}{2}}, \quad g_F = \left(\frac{\alpha^2 E_r F}{R_r l}\right)^{-\frac{1}{2}}, \quad H = \frac{h_{min}}{R_r}(\alpha E_r)^2$$

whereas

— · — · — lubrication subregime boundary

------------- pressure spike treshold for rolling contacts

Fig. 1 Dimensionless film thickness chart for the steady state case, from [11].

However, in reality many concentrated contacts have variations in load, tangential surface velocity, or curvature, with respect to time. This may result in a non-stationary behaviour of the lubricated contact, but it is not necessarily so.
The designation 'non-stationary behaviour' needs some more attention. In full film lubrication, with Reynolds' equation in force, a contact is defined to be non-stationary, if the so-called squeeze effect can no longer be neglected with respect to the wedge effect. Hence, the contact is called a dynamically loaded contact.
Well-known are dynamically loaded journal bearings, as an exponent of conformal contacts ([12], [13] and [14]).
In this paper counterformal contacts will be considered only, hence they are called dynamically loaded concentrated contacts if squeeze effects become important.

Moreover, in reality many concentrated contacts work in the mixed film region, due to a non-zero surface roughness, thus complicating the behaviour. Because the fluid film can be an extremely stiff element in a concentrated contact, full film considerations can still yield design directions if the contacting asperities share only a small portion of the total load. In the experiments reported in this paper, full film conditions exist.
The preliminary test results presented in this paper are part of an analysis of the lubrication of dynamically loaded concentrated contacts. The development of a theory for these contacts is in progress and will be reported later.

2 MEASUREMENTS IN LUBRICATED CONCENTRATED CONTACTS

The subject of this section is the measuring technique used in the experiments.

2.1 Dynamically loaded concentrated contacts

The question, "when is a concentrated contact with lubrication a dynamically loaded contact ?", is a difficult one to answer a priori. Most investigators assume that the behaviour is stationary, without verifying this assumption. Very often they are right in the concentrated contact case, because the entrainment velocity is high and the contact length is short, so a lubricant molecule will hardly 'feel' any variation during its passage through the contact. Gu [15], who presents a qualitative criterion for a steady state analysis, calls this passage time the contact zone flow time. He considers a contact as stationary, if the contact zone flow time is much smaller than the contact duration time. This idea is elaborated for a pair of contacting teeth from gears, and in this case the contact duration time equals the time for the contact to travel through the meshing cycle.

However, engineering practice provides some cases where a quasi-stationary approach is definitely in error, and situations in between. Examples of the former can be found in [16] and [17]. Examples of the latter one are very difficult to unmask and are the subject of these first experiments. References [16] and [17] deal with cam-follower contacts, where momentarily pure squeezing action occurs due to a zero entrainment velocity, hence a zero wedging action.

2.2 Vapour deposited transducers for lubrication studies

This section presents a very brief review of the development in vapour deposited transducers for pressure and temperature measurements in concentrated contacts with lubrication. Film thickness experiments will be reported later. Section 2.3 concerns some ground rules for transducers and deals with shape, dimensions, material and the manufacturing of the transducers.

Crook [18] was the first to employ a vapour deposited transducer in thin film lubrication. He used an evaporated chromium electrode on a glass disk to obtain film thickness readings on a 2 disk machine. About 3 years later, both Kannel and co-workers [19], [20] and Orcutt [21] showed pressure and temperature distributions in EHD contacts measured by vapour deposited microtransducers. Kannel is the first to observe the famous pressure spike in an experiment, and to use a real engineering surface for this transducer, viz., A.I.S.I. 52100 steel. At that time, transducer dimensions were about 25-50 μm wide (in rolling direction), 0.1 μm thick and several thousands of μm's long.

In Europe, Schouten [22] succeeded in precise measurements of temperature and pressure on a 2 disk machine, which were repeated at several institutes in Germany ([23], [24] and [25]). The smallest transducers are reported by Safa et al [26]; their manganin pressure transducer is manufactured by means of a mechanical mask and is only about 2.5 μm wide. In Poland, Janczak and co-workers have been using vapour deposited transducers for some time ([27]).

In all these references, the conditions are steady state with zero or moderate slip, with the exception of [24], where vapour deposited transducers are used in (conformal) dynamically loaded bearings, i.e., thick film lubrication.
The preliminary results presented here concern (a) moderate loads, thin film lubrication, unsteady state, pure sliding, and (b) high loads, thin film lubrication, quasi-steady state and low slip, respectively.

2.3 Thin film microtransducers for pressure and temperature measurements

To obtain contact pressure and temperature signals from a concentrated contact, a transducer must meet some stringent requirements. The most important requirements include:

1. high resolving power
2. fast transient response time
3. small interference with substrate surface topography
4. small perturbation of stress and temperature patterns on the substrate surface
5. high sensitivity to changes in the parameters to be measured, low sensitivity to other variations.
6. applicability on heat treatable steels.

A technique meeting these stringent requirements involves the use of vapour deposited transducers. Fig. 2 shows a schematic drawing of a thin film transducer, and provides some dimensions.
In general, these microtransducers can meet requirements (1)-(4a). By using an insulating layer of Al_2O_3, temperature transducers can fulfill condition (4t). Requirements (5) and (6) can be met by an appropriate choice of materials, e.g. Ti for temperature transducers, and manganin or NiCr for pressure transducers. In the measurements reported in this paper, NiCr was preferred to manganin, because its inherent better sensitivity to pressure variations and better adhesive wear properties (see also [24]).

By positioning the conductor pattern in the contact area, the transducer length can be chosen independently of the contact width, thus enabling arbitrary values of the electrical resistance. Owing to the processing equipment a value of about 1kΩ seems appropriate. The conductor pattern is made of Cu, Al or Au, because of their low electrical resistance. A small pattern area was chosen to achieve a low parasitic capacitance and to stay away

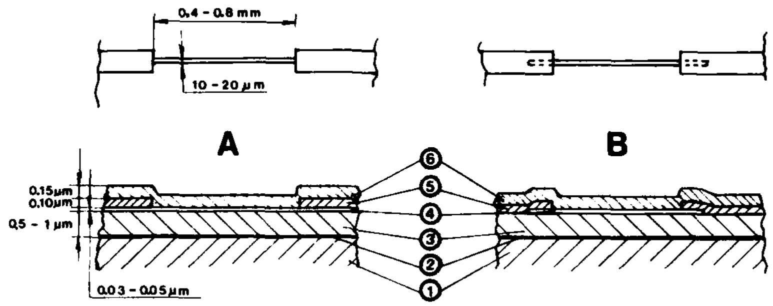


Fig. 2 Pictorial drawing of a thin film transducer, A: photolithographic; B: mechanical mask. (1) substrate; (2) adhesive layer; (3) insulating layer; (4) transducer; (5) conductor pattern; (6) protective layer (optional).

from pinholes. The terminals of the conductor pattern are connected to a print board by 20 μm dia. gold wires. A good bond is achieved by thermocompression. A protective coating may be applied on top of the completed transducer, e.g. when mixed film lubrication conditions predominate.
In manufacturing these transducers, two processes were developed:

1 Vapour deposition through a mechanical mask. This mask is made by applying a photolithographic technique on a 20 μm metal foil sheet. To obtain better adhesion, all layers should be evaporated without interrupting the vacuum. A carrousel for 4 masks was designed and built, allowing the manufacture of a complete transducer on the outer ring of a cylindrical roller bearing (see section 3), in one process run.

2 Vapour deposition or sputtering, followed by a photolithographic process. All layers are deposited or sputtered successively in one sequence, without breaking the vacuum. Afterwards, a selective etching technique permits the completion of the conductor and the transducer patterns. Finally, a protective coating can be deposited if desired.

Fig. 3 shows an example of the geometry of the transducers used in the experiments of section 4. The photolithographic process (2) was employed in the case of transducers for a cam-follower contact, and in the case of temperature transducers for a roller-ring contact, whereas the mechanical mask technique (1) was used in the case of pressure transducers for a roller-ring contact; see Fig. 4

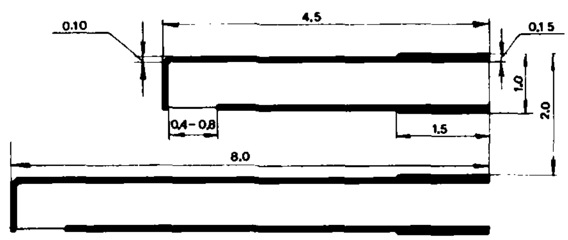


Fig. 3 Transducer geometry

The mechanical mask technique has a drawback if curved surfaces are to be provided with microtransducers, in that a different carrousel is needed for each different carvature. This problem is avoided by adopting photolithography, because the flexible photo masks will easily accommodate to the substrate curvature. In addition, photolithography enables the manufacture of complicated transducer shapes and, if needed, a large number of them on the same substrate.

2.4 Signal processing

When small changes in electrical resistance are to be measured (e.g., strain gages), often a Wheatstone bridge or a compensation circuit is used. In the present case a new electronic design by Bolk [28] was utilized. This design features a constant electric current flowing *through the transducer, having exactly the same value as the current through an ohmic refe-*

rence resistance. This so-called counter current circuit has an output voltage which is proportional to the change in the transducer's resistance, even at larger changes. This signal is then amplified by a broadband instrumentation amplifier (50x). Fig. 5 shows the circuit used throughout the experiments.

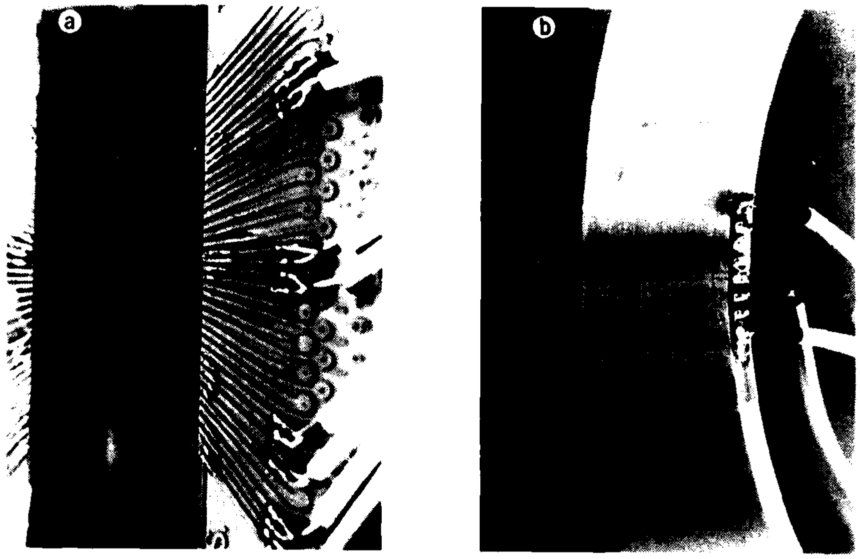


Fig. 4 a Flat follower plate with 40 transducers; photolithographic process
b Roller bearing outer ring with 4 transducers; mechanical mask technique

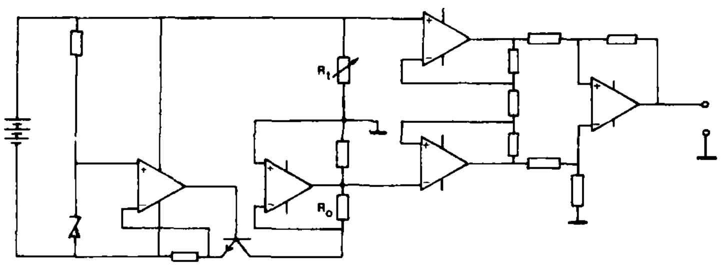


Fig. 5 Counter current circuit, followed by a broadband amplifier.
R_t: *transducer resistance*; R_o: *reference resistance.*

By using high-quality, low noise operational amplifiers with a large bandwidth and a high slew rate, this circuit can handle signals with frequencies up to 2 MHz. The high frequency output signal is stored on a digital recorder and a storage oscilloscope for further processing.

3 TEST RIGS FOR PRESSURE AND TEMPERATURE MEASUREMENTS IN CONCENTRATED CONTACTS

Two test rigs have been developed for pressure and temperature measurements in concentrated contacts of machine elements:

1 a cam and flat follower machine
2 a radial cylindrical roller bearing test apparatus.

For both measuring devices a design is chosen where an "array" of thin film transducers is deposited on one of the contacting surfaces. In addition, it is also possible to perform measurements in the sliding areas of these contacts, using only one transducer. In that case the transducer is moved stepwise through the loaded area during operation of the test rig.

3.1 Cam and flat follower machine

Vichard and Godet [29] are the first investigators who have performed experiments on a cam and flat follower test rig. However, they were only interested in the global film thickness in the contact, which was measured by a capacitance technique.

Fig. 6 shows the cam and flat follower test rig. The horizontal camshaft (1) forces a piston-like follower body (2) to move in a horizontal plane. This follower is supported by a double hydrostatic bearing (3). The load in the cam-follower contact is measured by means of a load cell (4). The complete unit, which includes follower, hydrostatic bearing and load cell, can be moved in a vertical plane using a linear bearing. The transducers are evaporated on a flat follower plate.

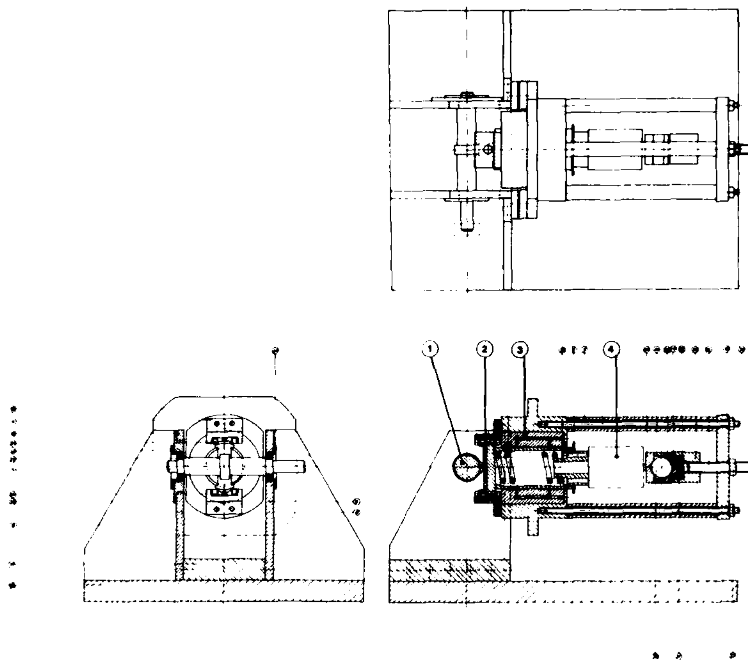


Fig. 6 Cam and flat follower test rig. See text for explanation of numbers 1-4.

As a result of the translatory vertical motion, the sliding area of the cam-follower contact is moved. Hence, the transducer location relative to the nominal contact position orbit is changed. Thus it is possible to position a single transducer in any specified location. During normal operation, the contact passes the transducer two times per revolution of the camshaft, see fig. 7, if it is located in the sliding area. In general, these two contact situations correspond to two different specific positions on the cam circumference. The measuring procedure is described in [30]. Fig. 8 shows the total test arrangement. In fig. 9 the cam-follower contact is shown in more detail, with 2 arrays of 20 transducers on the follower plate.

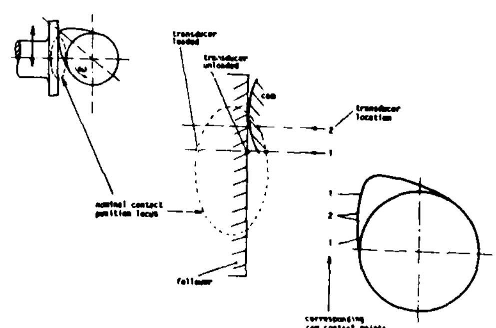


Fig. 7 Measurements in a cam-follower contact.

Fig. 8 Cam and flat follower test rig.

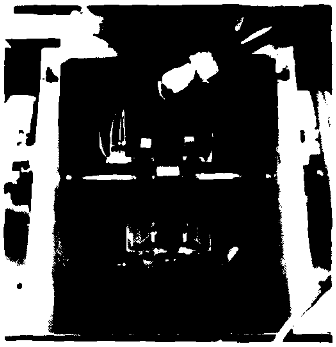


Fig. 9 Cam-follower contact showing transducers mounted.

3.2 Radial roller bearing test rig

A drawing of the test apparatus for radial cylindrical roller bearings is shown in fig. 10. *Test bearing* (1) is loaded by a radial tensional force, which is measured by load cell (2). The test bearing is mounted in a self-aligning device (3) to ensure proper line contact between rollers and ring surfaces. Two possibilities can be used to locate the transducers: the rotating inner ring, or the stationary outer ring. In this case, the *outer ring* was chosen, thus obviating a rotating contact. The outer ring can be rotated stepwise over small fixed angles, thus allowing *measurements* at *different* transducer locations. See figs. 11 and 12.

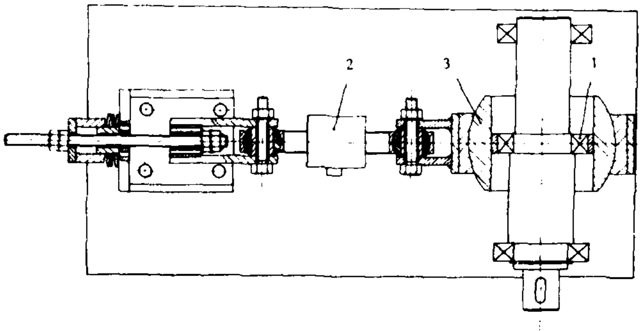


Fig. 10 Radial roller bearing test rig.
See text for explanation of numbers 1-3.

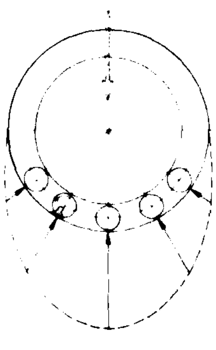


Fig. 11 *Measuring principle.*

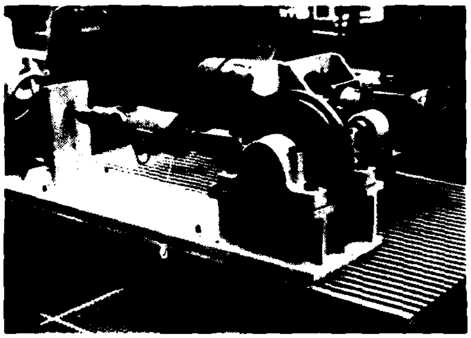


Fig. 12 Radial roller bearing test rig.

4 RESULTS

When thinking out an experiment under EHD conditions, it is necessary to define the subregime of full film lubrication of the contact. If steady state conditions prevail, Johnson's famous map of the subregimes of full film lubrication can be used ([31]). However, this map consists of dimensionless numbers, which all contain load and speed. As these two variables may change during an experiment (and, in general, not the elasticity of the surfaces or the pressure-viscosity coefficient of the lubricant), the map from [11] is more appropriate. Moreover, lines of constant relative film thickness Λ coincide with lines of constant film thickness for a given contact, permitting an indication of the mixed film / full film lubrication transition at about $\Lambda = 1$. These tresholds are marked in fig. 13 Λ_a and Λ_b, for the cam- flat follower, and the roller bearing test rig, respectively. The hatched rectangle in fig. 13 designates the area investigated by Dowson and Higginson ([4], p. 101). In addition, the test conditions are marked in this figure. Hence, from a steady state point of view, the experiments are run in the full film EHD lubrication regime.

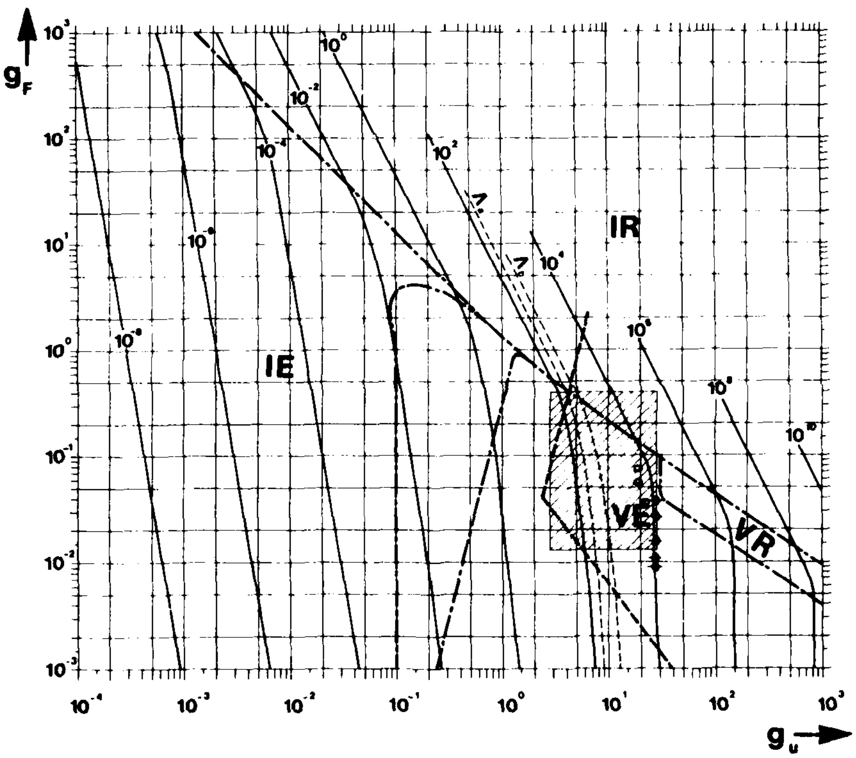


Fig. 13 Dimensionless film thickness chart, containing mixed film / full film lubrication transition tresholds: Λ_a (cam-follower) and Λ_b (roller bearing), and conditions at experiments: o (cam-follower) and + (roller bearing). The hatched area designates the Dowson and Higginson [4] investigations.

4.1 Cam and flat follower machine results

Some data: $E_r = 2.3 * 10^{11}$ N/m²
$\eta_0 = 0.747$ Ns/m² (27°C)
$\alpha = 2.89 * 10^{-8}$ m²/N (27°C)
$m = 2.24$ kg
$c = 7.93 * 10^4$ N/m
$l = 17.5 * 10^{-3}$ m
$R_r = 30.49 * 10^{-3}$ m
$e = 4.09 * 10^{-3}$ m } eccentric cam

In these preliminary experiments an eccentric cam with $\sigma_{cam} = 0.12 * 10^{-6}$ m (RMS) was used. The surface finish of the follower is much better, *and can be neglected in the calculation* of Λ_a. Hence, $H_{crit\ a} \approx 200$.

For a preload (at zero follower lift) of 500 N and a speed of 10 rev/s fig. 14 was obtained. In this figure the following scales were used: (1) vertically, in fig. 14A, for the pressure signal p: *0.073 GPa/div, and for the temperature signals* T (top): 0.36°C/div, T(bottom): 3.04°C/div, respectively, and in both fig. 14B and fig. 14C, for the pressure signals p: 0.035 GPa/div, and for the temperature signals T: 3.0°C/div, and (2) horizontally, for the time reference t: $1.09 * 10^{-3}$ sec/div. The reproducibility of both signals is good, in particular of the temperature transducer.

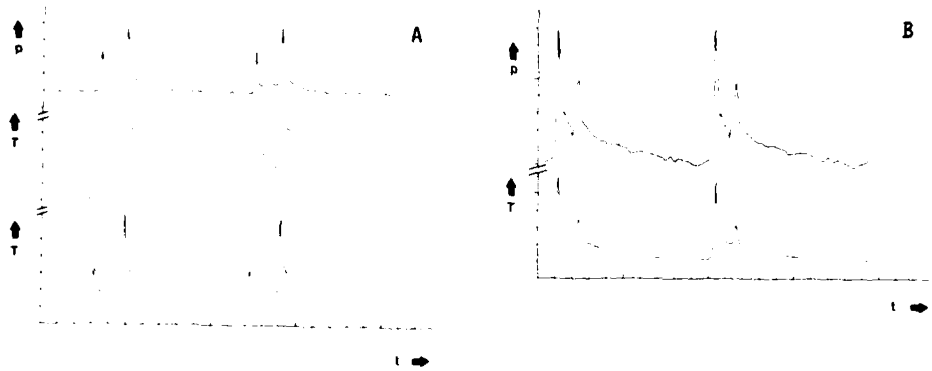
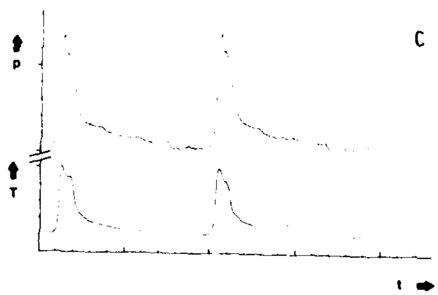


Fig. 14 **Pressure and temperature measurements in a cam-follower contact.**
Transducer position:
A : $3 * 10^{-3}$ m before maximum lift
B : $3.8 * 10^{-3}$ m after maximum lift
C : $4.1 * 10^{-3}$ m after maximum lift
See text for more details.



4.2 Radial roller bearing test rig results

Some data: $E_r = 2.3 * 10^{11}$ N/m²
$\eta_0 = 0.610$ Ns/m² (30°C)
$\alpha = 2.81 * 10^{-8}$ m²/N (30°)
$R_r = 8.315 * 10^{-3}$ m
$l = 13 * 10^{-3}$ m

The surfaces in the outer ring contact have a composed RMS surface roughness value of *about* $\sigma_{roller/ring} = 0.12 * 10^{-6}$ m, which results in a mixed film/full film lubrication transition at about $H_{crit\ b} \approx 500$, due to the larger curvature of this contact compared to the cam-follower contact of section 4.1.

For a speed of 540 rpm and static loads of 500 N, 1,000 N, 2,500 N, 5,000 N, and 7,500 N, *the results of Fig.* 15A *and* 15 B *were obtained. In this figure the following* scales were used: (1) vertically, for the temperature signals T: in fig. 15A:

0.5°C/div, 0.5°C/div, and 1.25°C/divat 500N, 1,000N, and 5,000N, respectively, and in fig.15B: 0.25°C/div, 0.5°C/div, 1.25°C/div, 1.25°C/div, and 2.5°C/div at 500N, 1,000N, 2,500N, 5,000N, and 7,500N, respectively, and (2) horizontally, for the time reference t: in fig. 15A: $1.09*10^{-2}$ sec/div, and in fig. 15B: $2.18*10^{-3}$ sec/div. As can be seen from fig. 15A, the reproducibility of the temperature signal is poor.
Pressure measurements in the contact are in progress and will be reported later.

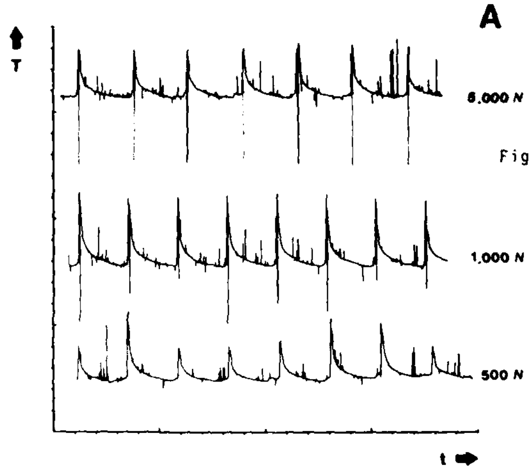
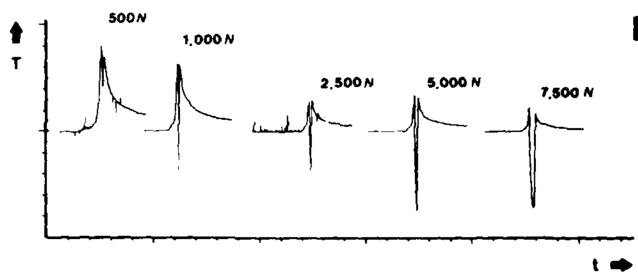


Fig. 15 Temperature measurements in a cylindrical roller/outer bearing ring contact, at 540 rpm. Transducer positioned at maximum load.
See text for more details.



5 DISCUSSION

From Fig.14 it can be concluded that temperature variations in a cam-follower contact can be surprisingly small, of the order of 10 degrees Centigrade. Previously, the existence of a full film was confirmed by means of a global capacitance technique. The increase in temperature never exceeded 15°C, except when mixed film lubrication conditions prevailed. These conditions can easily be introduced by a slight misalignment of the cam-follower contact, or at very low speeds. The upper temperature curve of Fig. 14a was supposedly obtained under the lee of the mixed film lubricated side of the contact, because the maximum temperature rise is only 1.6°C, and wear scars were observed at the opposite side of the transducer employed in this experiment (see also Fig. 4a). Under mixed film conditions temperature variations up to 50°C at 0.37 GPa were found.

The transducers in the cam-follower contact were used at pressures up to 0.37 GPa.
If a correct alignment can be realized, these transducers can work well for more than 10 hours, which suffices for our purposes.

In the near future a different cam geometry will be used, allowing loads up to 0.65 GPa and momentarily zero entrainment velocity. This will be the subject of a forthcoming paper.

From Fig. 15 it follows that the maximum temperature rise detected is about 3°C at 7,500 N. Fig. 15B suggests a temperature collapse around the expected maximum at high loads, which is highly suspect. At high loads, the duration of this sharp drop amounts 2 Hertzian contact widths plus 1 conductor pattern track. Thus, the rollers probably shortcut the transducer during each overrolling. The protective layer did not show any wear after the experiments. Due to its small thickness, pinholes are most likely to be responsible for these shortcuts. But then, the origin of the phenomenon lies in the lubrication regime of the contact. It should be full film lubrication, see Fig. 13, but probably mixed film conditions abound. This viewpoint is supported by Fig. 15A, which shows that there are no identical temperature profiles. Each contact operates under different conditions. This can be due to a deficiency in the test rig, or to manufacturing tolerances in the bearing components.

Consequently, the temperature rise can be higher than 3°C, up to about 10°C. It should be kept in mind that in this case the lubrication mode is mixed film. Under normal running conditions temperature extrema can be much lower. Even then, the resolution power of the microtransducers is sufficient.

The maximum pressure at the roller bearing experiments was about 0.97 GPa. After running our tests, which took about 20 hours, only one transducer failed out of 5.

It is necessary to verify the lubrication condition in the roller-ring contact. This can be done by resistance or capacitance measurements over the contact. These measurements will be carried out soon.

6 CONCLUSIONS

Vapour deposited transducers were manufactured and tested under adverse conditions. A manufacturing technique, the photolithographic process, was introduced to make production easier and to improve the experimental results. The transducers can be made sufficiently small and have a satisfying life expectancy. Signal processing by a counter current circuit worked properly. Measurements will be extended to higher loads and more unsteady conditions.

ACKNOWLEDGMENTS

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WEAR OF HIGH SPEED ROLLER BEARINGS

by

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SUMMARY

A laboratory bearing test rig was constructed to study wear in three high speed roller bearings simultaneously under different conditions of loading and lubrication. A test run of 355 hours duration was performed during which wear of the bearings was monitored using DR ferrography. The test bearings from the rig were examined using metallography and microhardness measurements. Wear debris particles were collected from oil samples using ferrography. These were studied using bichromatic microscopy, scanning electron microscopy and energy dispersive X-ray analysis. The results of ferrography were correlated with the metallographic results.

1. INTRODUCTION

Failure of rolling-element bearings may result from a number of different reasons, some of which are attributable to material defects, rolling contact fatigue or wear. Bearing failures due to material defects have been greatly reduced by using clean (vacuum melted and degassed) and properly heat-treated steels. Rolling contact fatigue can cause surface and subsurface initiated failures of the bearings under extreme stress conditions. Subsurface microstructural changes due to rolling contact fatigue may occur in the bearing components if the maximum shear stress exceeds a threshold value [1]. These changes may greatly reduce the fatigue life of the bearing.

Wear of the bearings results from metal-to-metal sliding due to inadequate lubrication or ingress of abrasive particles. Under inadequate lubrication, structural changes may occur in the surface material of bearing components as a result of rolling and sliding action. These changes result in the formation of friction layers [2] at the rubbing surfaces. It was shown [3] that formation of friction layers on the surfaces of high-speed roller bearings is very detrimental, because they can upset the hardness balance between bearing components and cause excessive wear leading to the failure of the bearings even under light loadings.

Laboratory simulated wear tests may provide useful information to understand wear mechanisms and failure modes of the bearings in actual engine applications. The purpose of this paper is to report on the development of a laboratory test rig and to describe the observations made on three bearings used during a test run.

2. BEARING TEST RIG

A bearing test rig was built to test wear performance of three high-speed roller bearings simultaneously. The basic components of the rig are shown in Fig. 1. These include a shaft assembly, a belt drive system, three bearing housing assemblies numbered 1, 2 and 3, respectively from the belt end of the shaft, lubrication supply lines and thrust bearings to prevent axial movement of the shaft.

The inner rings of the bearings are press fitted on to the shaft. The shaft is made of AISI 4140 steel which was through hardened to 50 HRC and centre-ground to a surface finish of approximately a 0.3μm r.m.s. The straightness of the shaft was found to be

critical for successful operation of the rig and therefore the shaft is ground true within 80μm.

The shaft is supported by the three test bearings, while simple thrust bearings prevent axial movement. Each thrust bearing consists of a hardened steel ball and a friction plate polished to minimize wear of the ball.

Each test bearing support consists of three components: the bearing housing, the base and base clamp.

The shaft is rotated by a belt attached to the pulley of the drive system. The drive system consists of a 15 hp (11 kW) AC electric motor running at a speed of 1750 rev min^{-1}. A speed ratio of 15 gives a rotational shaft speed of 25000 to 26000 rev min^{-1}. The drive system imposes a radial load on the bearings through the shaft due to belt pull. However the bearings can be further loaded by shimming one of the bearing housings.

The lubrication system contains an oil tank, an electrically driven oil pump, pressure regulating and flow valves, pressure gauges, a flow meter, an oil filter and a cooling coil.

3. EXPERIMENTAL PROCEDURE

3.1 Test Conditions

In addition to the load on the bearings due to belt pull the bearings were further loaded by shimming the No. 3 bearing housing. To determine the static loading on the bearings a calibration procedure was used. This involved using strain gauges to measure shaft deflections produced by applying known loads to the No. 2 bearing. The load applied to the shaft by the belt was also determined by measuring the shaft deflection. As a result two calibration curves (load versus strain) were produced to determine the load applied to the shaft by the belt and the load developed on No. 2 bearing due to both belt pull and shimming. This allowed the calculation of static loads produced on the other bearings (No. 1 and No. 3). The static loading on the bearings was gradually increased by increasing the shimming of the No. 3 bearing housing. The calculated static load values for each bearing in each test period are listed in Table 1. However dynamic loadings for this test were not determined.

The temperature of each bearing and of the lubricating oil was continuously monitored using type-T thermocouples and a digital thermometer. Thermocouples were attached to the outer ring of each bearing. In each test period, temperatures of the bearings and the lubricating oil stabilized in approximately two hours of operation, following a sharp initial increase. Stabilized temperatures (after five hours of continuous operation) of the bearings and the lubricating oil are listed in Table 1.

Rotational speeds of the shaft and the cage of No. 2 bearing were measured using fibre optic light guides. The signal from the fibre light guide was amplified and displayed on the screen of an oscilloscope in the form of a square wave. The frequency of the wave corresponding to the rotational speed of either shaft or cage was read on a frequency counter. The rotational speeds of both the shaft and the bearing cage were also recorded by a computer which was used to monitor the test conditions (See Table 1).

The test rig was operated for approximately 10 hours every day. For safe operation, test variables were monitored continuously by the computer. If shaft or cage speeds or axial movement of the shaft exceeded predetermined limits the computer would switch off the rig. Another safety device was installed to switch off the rig in case of wandering of the drive belt.

Each bearing was separately lubricated with synthetic oil meeting MIL-L-23699C specification, under a constant pressure selected within a range of 280 to 560 kPa. The flow rate of the oil for each bearing was also separately selected and maintained reasonably constant for each test period, as shown in Table 1. During the first 150 hours the oil flow rates to the bearings were gradually decreased as shown in Table 1. After 150 hours of operation the No. 3 bearing was run dry. A 5μm oil filter was placed in the oil supply line, so that the oil was continuously cleaned before it reached the bearings.

Oil samples were collected from each bearing regularly at 5 hour intervals. The samples were subsequently analysed by direct reading (DR) ferrography to monitor the wear behaviour of the test bearings. To collect wear debris samples from the No. 3 bearing during the dry run period, the bearing was washed briefly at every 5 hour period.

The No. 2 bearing failed just after 270 hours of test following a period of noisy operation and it was replaced with a new bearing to continue the test run. The test run was terminated at 355 hours due to a shaft failure.

3.2 Metallography

The surfaces of the bearing components were cleaned and electroplated with nickel for edge retention. Each specimen was prepared using standard metallographic techniques

Table 1. Data for the bearing test run

Test time (hours)	Oil flow rate to bearings (Litre/min)			Static load on the bearings (N)±100N			Temperature after 5 hours (°C), ±2°C				Shaft speed	Cage speed
	No. 1	No. 2	No. 3	No. 1	No. 2	No. 3	No. 1	No. 2	No. 3	Toil	(rev min^{-1})	(rev min^{-1})
0-50	3	3	3	1972	1864	755	83.6	81.6	81.7	73.5	25620	10440
50-100	3	3	3	2021	1991	834	84.9	83.2	84.8	76.5	25620	10440
100-120	1	1	1	2021	1991	834	89.1	86.3	85.8	78.6	25620	10440
120-140	1	0.5	0.5	2021	1991	834	87.6	89.5	89.1	76.2	25680	10400
140-150	1	0.5	0.2	2021	1991	834	87.6	90.7	103.0	72.3	25710	10440
150-170	1	0.5	0	2021	1991	834	85.7	86.5	97.0	71.3	25740	10440
170-250	1	0.5	0	2040	2060	873	85.5	88.5	97.3	75.1	25740	10440
250-355	1	0.5	0	2060	2109	903	87.1	90.1	106.1	76.0	25710	10440

and etched in 2% Nital, followed by examination with both optical and scanning-electron microscopy (SEM) with energy dispersive X-ray facilities.

Microhardness measurements were made on the polished and lightly etched sections of the bearing components using a Tukon tester with a Knoop indenter.

3.3 Ferrography

Oil samples collected from each bearing during the test were analysed by DR ferrography and the results are shown in Fig. 2. In order to study wear particles, ferrograms were made from the oil samples collected from the bearings. The ferrograms were examined under an optical microscope using either bichromatic illumination or reflected light. The ferrograms were further studied in an SEM and the wear particles were identified using energy-dispersive X-ray analysis.

4. RESULTS

4.1 New Bearing

The dimensions of the test bearing are shown in Fig. 3. Three major components (rollers, inner ring and outer ring) of the bearings were produced from AISI 52100 steel, the chemical composition of which is given in Table 2.

The microstructure of the bearing steel consisted of speroidal carbides, $(Fe,Cr)_3C$, homogeneously distributed in a matrix of tempered martensite, as shown in Fig. 4. X-ray diffraction studies showed no observable retained austenite in the steel.

The hardness of the bearing components were found to be in the range of 60 HRC, which met the normal engineering requirements [4].

The bearing cage was made of leaded high-strength bronze and plated with silver.

Table 2. Chemical composition of bearing steel, wt.-%

C	Si	Mn	S	Cr	Al	Fe
1.0	0.34	0.32	0.02	1.60	0.03	balance

4.2 No. 1 Test Bearing

SEM examination showed that the roller surfaces of No. 1 bearing were characterized by a network of pits, as shown in Fig. 5. The inner and outer raceways of this bearing also showed pits together with spalls and scoring marks as shown in Figs. 6 and 7, respectively. However, roller surfaces showed more pitting than either raceway.

No distinct microstructural changes were observed in the surface material of the bearing components. Fig. 8 shows the typical microstructure of the surface region of the rollers.

No hardness change was found in the surface material of the bearing components up to 5µm depth from the surface.

Fig. 9 shows the entry deposit of a ferrogram made from the oil sample collected

from No. 1 bearing at 270 hours. The entry deposits show strings of tightly joined magnetic wear particles. Although several particles appeared in different colours under reflected light, energy dispersive X-ray analysis showed that they were all from the bearing steel. It can be seen that the particles are very fine (ranging from 1 to 7μm in size) and mainly flat, indicating a normal wear condition. However the ferrogram, Fig. 10, obtained at 350 hours from this bearing showed a change in the shape and size of the wear particles. The particles became more equiaxed and their sizes are slightly increased indicating a transition from mild wear to more severe wear. A ferrogram made from the oil sample collected from this bearing at 355 hours just after the shaft failure, showed spherical and cutting wear particles, as shown in Fig. 11. Spherical particles which are characteristic of rolling contact fatigue were also observed in the ferrogram of this bearing obtained at 350 hours. However the cutting wear particles are not typical deposits of this ferrogram and were only observed in this particular area. Energy dispersive X-ray microanalysis showed that some of the cutting wear particles are nickel. Since there is no nickel part in the test rig, the nickel or possibly all cutting wear particles are presumed to be from a foreign source, most likely picked up during preparation of the ferrogram.

4.3 No. 2 Bearing

Fig. 12 shows a photograph of No. 2 bearing which failed at 270 hours. It can be seen that all the components of the bearing including the cage were damaged. It was observed that the inner ring of this bearing had moved on the shaft, away from the bearing, causing a noisy operation just before the failure. This was due to the axial loading produced on the bearing by greater shaft deflection which resulted from increased shimming.

Examination of the roller surfaces with the SEM showed fractures and spalls, as shown in Fig. 13. However some surface areas appeared to be smooth and polished. Fig. 14 shows the details of the spalling produced on the inner ring raceway. The outer ring raceway of this bearing showed more surface damage than the inner race, as some areas were almost covered by spalls and scars, shown in Fig. 15. It can be seen that some areas of the outer raceway rib were also fractured and broken. These observations suggest that surface damages (fractures and large spalls) on the bearing components resulted from overstressing developed due to combination of both radial and axial loading during operation. Therefore the cause of failure was not excessive wear, but improper loading developed on this bearing during operation.

Examination of the surface material of the bearing components showed no observable microstructural changes. A typical surface microstructure of the rollers is shown in Fig. 16.

Fig. 17 shows the heavy entry deposits of a ferrogram made from the oil sample collected from this bearing at 270 hours just before the failure. Energy dispersive X-ray analysis showed that all the debris particles are from the steel components of the bearing. Another SEM micrograph, Fig. 18, shows strings of the tightly joined wear particles from the bearing steel, in an area just below the entry deposit of the same ferrogram. The density and the size of magnetic particles decreased with increasing distance away from the entry deposit. However a few free large particles were observed below the entry deposit of this ferrogram. A typical example of this is shown in Fig. 19. X-ray microanalysis showed that the large particles are from the cage of the bearing, while strings of finer particles are from the steel components of the bearing. These observations clearly show the abnormal failure situation of the bearing, because of large debris particles from the bearing steel and the presence of particles from the cage indicating cage failure.

4.4 No. 3 Bearing

SEM examination showed that the rollers of No. 3 bearing were almost covered by pits, as shown in Fig. 20. Again pitting was found to be the major surface deterioration for the inner ring raceway of this bearing. The pits, on the inner raceway surface, were rectangular and showed an orientation in a direction approximately 72^{o} away from the rolling direction as shown in Fig. 21. The outer raceway surface appeared to be smoother with fewer pits and scoring marks. However in some areas near the edge of the roller path along the rib sides more surface pitting and scoring marks were observed as shown in Fig. 22. This observation suggests that the rollers were tilted during operation and the roller ends (corners) came into contact with both raceways causing more surface damage (pitting and scoring) along the edge of the roller path.

Optical microscopy revealed light and dark etching regions (LER and DER) in the surface material of the roller ends as shown in Fig. 23. An SEM micrograph, Fig. 24, shows the detail of these different etching regions. The matrix of the light etching region (LER) showed very little structural detail even after deep etching. However the dark etching region (DER) showed a coarser microstructure than that of the new bearing steel. The measured microhardnesses of the different etching regions are given in Table 3. It can be seen that the LER is harder than the unaffected bearing steel, while the DER is the softest constituent.

The etching characteristics and hardness of the LER suggest that it may be martensite caused by momentary excursions into γ or $\gamma + M_3C$ phase fields as suggested by previous workers [2,3] The DER would then be overtempered martensite derived from the original microstructure.

Table 3. Average microhardness values of different etching regions

Region	Microhardness, Knoop
LER	860
DER	500
Unaffected region	720

However these different etching regions were not observed in other surface areas of the rollers. Some cracks were observed in the surface material of the rollers. Fig. 25 shows a crack which appears vertical to the surface on the longitudinal section of a roller. No inclusions were found to be associated with these cracks and the observations suggest that the cracks were initiated from the surfaces and propagated into the roller.

Examination of the surface material of the bearing components at higher magnifications showed no distinct microstructural changes. A typical surface microstructure of the rollers is shown in Fig. 26.

A ferrogram, made from the oil sample collected from this bearing at 350 hours showed strings of magnetic wear particles at the entry deposit, as shown in Fig. 27. Energy dispersive X-ray analysis showed that the wear particles in the strings were from the bearing steel. A larger particle identified as silver from the plating of the bearing cage is marked.

5. DISCUSSION

The No. 1 bearing showed very little wear, the No. 2 bearing failed completely at 270 hours, and the No. 3 bearing showed considerable wear and microstructural changes particularly at the roller ends. In addition the cage of the No. 3 bearing was found to be stuck following the shaft failure at 355 hours.

These observations may be explained in terms of loading and lubrication conditions of the bearings. The No. 1 bearing was well lubricated throughout the test run, loading was mainly radial and the load level was less than the calculated radial load capacity (2330N) of the bearing. Therefore excessive wear in this bearing would not be expected. Although the radial loading on the No. 2 bearing was in the same range as that on the No. 1 bearing, it failed because axial loading caused movement of the inner ring during the run. The No. 2 bearing occupies a critical point on the shaft, because it constrains the shaft deflection produced by the belt pull and the shimming of the No. 3 bearing housing. This produces complex loadings on this bearing during rotation of the shaft. Simply put, shaft deflection applies an axial loading to the bearing which in this case lead to an observed movement of the inner ring along the shaft during the test run. Since the cylindrical roller bearings are designed to carry radial loads, excessive axial loading can cause bearing failure. Therefore the cause of failure of this bearing was not wear due to reduced lubrication, but improper loading (mainly axial) developed due to increased shimming. This case may represent an extreme operating condition for these bearings.

The No. 3 bearing was less heavily loaded, but was subjected to oil starvation after 150 hours of initial run. Examinations suggest that the axial loading produced on this bearing due to shimming was not high enough to cause failure, but it was sufficient to cause some roller tilting or skewing. As a result of oil starvation and roller skewing, the roller ends of this bearing came into contact with the raceways producing the friction layers referred to above as LER and DER. Similar friction layers were also observed on the components, particularly the roller ends, of a used bearing taken from an engine [3]. The LER was identified as friction martensite and the DER as overtempered martensite.

However no other distinct friction layers were observed in the surface material of the components of any of the test bearings. This indicates that the components of the bearings were not subjected to sliding with metal-to-metal contact. This is an expected result, because the No. 1 and No. 2 bearings were sufficiently well lubricated. Although the lubricating oil to the No. 3 bearing was cut off after 150 hours, the oil mist produced during the run provided some lubrication for this bearing. In addition, it was washed with oil every 5 hours to collect wear debris particles.

Metallographic examination showed that the rollers of the bearings suffered more surface pitting than either raceway. Surface pitting may result from fatigue even under ideal running conditions. However it is expected that the rollers were heated more than the raceways, because of their smaller mass.

The static radial loadings produced on the test bearings were higher than the estimated maximum dynamic load (580N) which they support in actual engine application. However no subsurface microstructural changes were observed in the components of the test bearings. It was reported that to produce stress-induced microstructural changes in this bearing steel, the alternating maximum shear stresses must exceed a threshold value of 724 MPa [1]. Thus, the maximum shear stresses produced in the components of the test

bearings were not high enough to cause subsurface microstructural changes. To obtain shear stresses greater than the threshold value of 724 MPa, in the components of the test bearings the radial loading on the bearings must exceed 3000 N. Therefore for radial loadings below approximately 3000N, subsurface rolling contact fatigue is not a significant wear or failure mechanism in these bearings. However another test run is in progress with increased loading.

The No. 2 bearing failed suddenly as a result of movement of the inner ring. Thus, ferrography, which enables one to monitor progressive changes, was unable to give sufficient warning of this failure. However ferrography gave some information to monitor wear in the other bearings such as the No. 3 bearing. Following a sharp increase in the quantity of wear particles from the No. 3 bearing at 350 hours, the bearing cage was found to be stuck at 355 hours. In the present study, the actual quantity of the wear particles in the oil samples may be affected by the sampling procedure. However examination of the wear debris particles deposited on the ferrograms provided useful information about operating wear mechanisms in the bearings. Small and flat wear debris particles, ranging from 0.5 to 5μm in size, indicated mild wear condition in all bearings up to 270 hours. A ferrogram of the oil sample collected from the No. 2 bearing at 270 hours just before the failure, showed very large wear debris particles (up to 50μm in size) from the steel components of the bearing together with large silver and bronze particles from the bearing cage. Therefore the presence of large particles from the bearing cage in the oil samples may indicate bearing failure.

A change was observed in the shape and the size of the wear particles collected from the No. 1 bearing after 270 hours. At 350 hours the wear particles of this bearing appeared more equiaxed and the average particle size was increased indicating a transition from mild to more severe wear condition. A ferrogram made from the oil sample of the No. 3 bearing at 350 hours showed rounded equiaxed wear particles from the bearing steel and silver particles from the bearing cage. The shape and size of the wear particles and the presence of silver particles indicated a severe wear condition. After 5 hours (at 355 hours) following the oil sample collection, the cage of this bearing was found to be stuck. Metallographic examination of this bearing after 355 hours of run confirmed the ferrographic results by revealing the severe wear on the rollers. Therefore presence of large silver particles in the lubricating oil indicates a severe wear situation.

Correlation of ferrographic results with the worn surface examinations is necessary to understand wear mechanisms operate in the bearings. However at this stage examination of wear debris particles is not complete. Metallographic observations suggest that the observed normal wear particles were produced by surface pitting of the bearings, while the large particles resulted from spalling and from the formation of hard and brittle martensitic layers particularly on the corners of the rollers. Examination of structure of wear particles with transmission electron microscopy (TEM) using electron diffraction may provide further information about the wear mechanisms in the bearings. Such work is in progress.

6. CONCLUSIONS

1. Development of axial loading on cylindrical high speed roller bearings is more detrimental than increased radial loading, and excess axial loading can cause sudden bearing failure.

2. Oil starvation is more detrimental to the life of the bearings than increased radial loading in the range tested. This is likely to be the case for radial load at least up to 3000 N for the test bearings.

3. The observed friction layers (LER and DER) on the ends of the rollers of the No. 3 bearing resulted from heat of friction due to metal-to-metal sliding.

4. Rollers of high speed bearings suffer more surface pitting than the raceways even under normal running conditions. However surface pitting and other phenomena relating to rolling contact fatigue are not significant wear or failure mechanisms in these bearings.

5. Small size, flat wear debris particles indicate a mild wear condition in these bearings, while the large and equiaxed particles indicate a severe wear mechanism.

6. Presence of large silver particles in the oil samples indicates a severe wear condition leading to failure, however presence of large bronze particles from the cage indicates bearing failure.

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8. ACKNOWLEDGEMENTS

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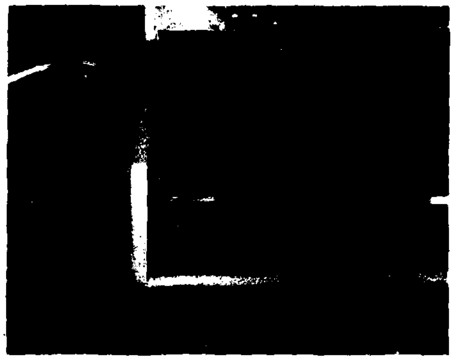

Fig. 1. Photograph of bearing test rig.

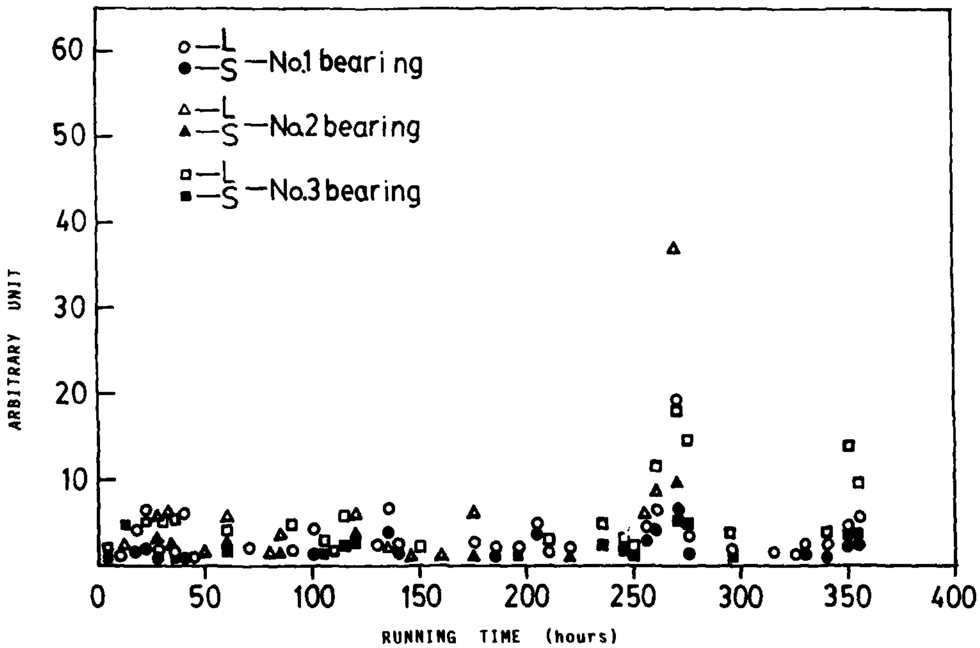

Fig. 2. Results of DR ferrography obtained from three bearings.

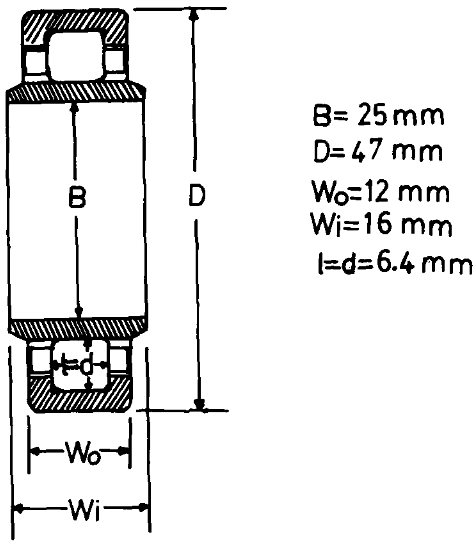

Fig. 3. Dimensions of test bearing.

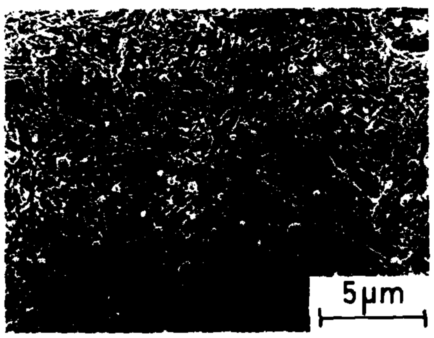

Fig. 4. Original microscructure of bearing.

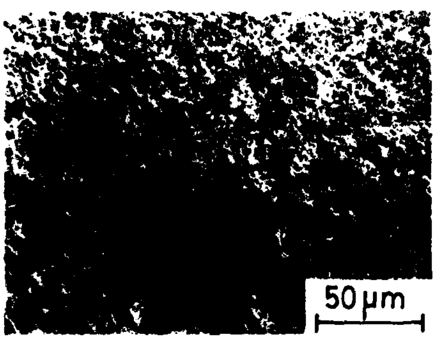

Fig. 5. Worn roller surface of No. 1 bearing.

Fig. 6. Inner ring raceway of No. 1 bearing.

Fig. 7. Outer ring raceway of No. 1 bearing.

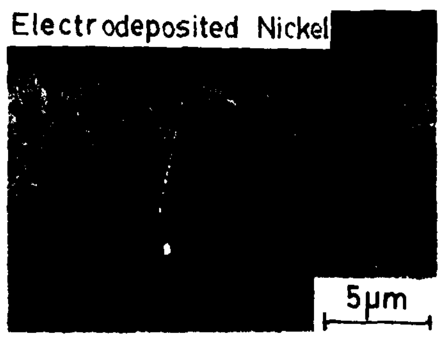

Fig. 8. Surface microstructure of a roller of No. 1 bearing.

Fig. 9. Wear debris particles collected from No. 1 bearing at 270 hours.

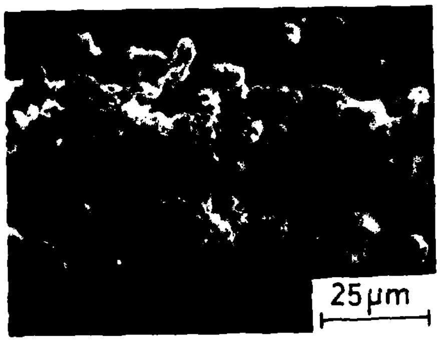

Fig. 10 Wear particles collected from No. 1 bearing at 350 hours.

Fig. 11. Spherical wear particles collected from No. 1 bearing at 355 hours.

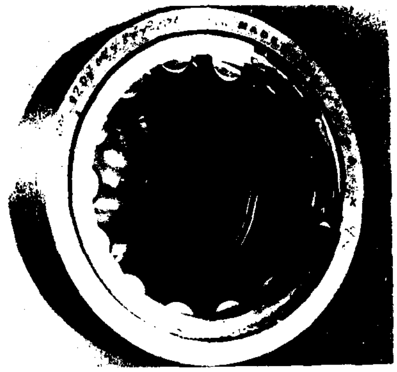

Fig. 12. Photograph of No. 2 bearing after failure.

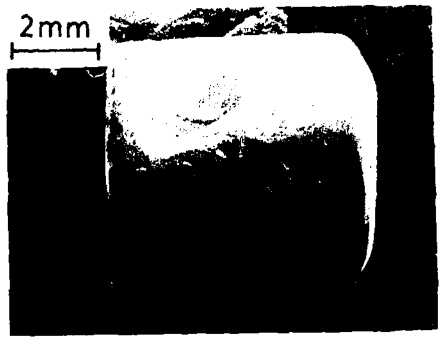

Fig. 13. A roller of No. 2 bearing after failure.

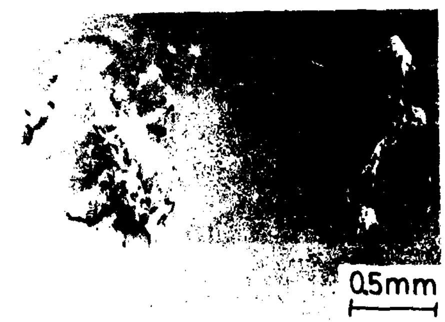

Fig. 14. Inner ring raceway of No. 2 bearing.

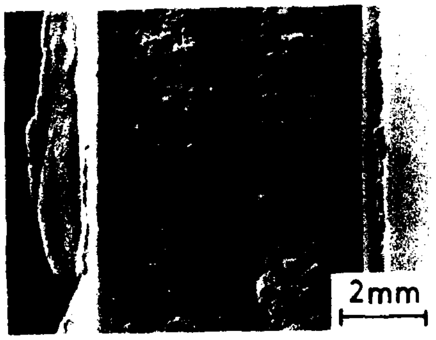

Fig. 15. Outer ring raceway of No. 2 bearing after failure.

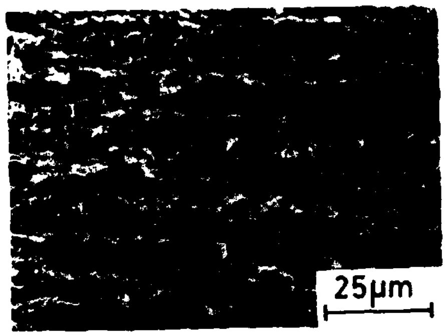

Fig. 18. Strings of wear particles of No. 2 bearing just below the entry deposit shown in Fig. 17.

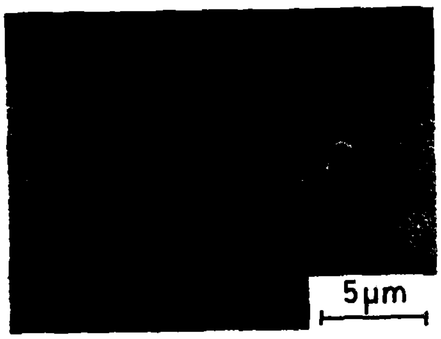

Fig. 16. Typical microstructure of rollers of No. 2 bearing.

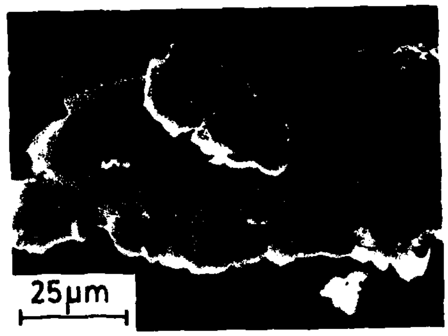

Fig. 19. Large bronze particles from the cage of No. 2 bearing observed below the entry deposit of a ferrogram obtained at 270 hours.

Fig. 17. Heavy entry deposits from a ferrogram made from oil sample of No. 2 bearing collected at 270 hours just before failure.

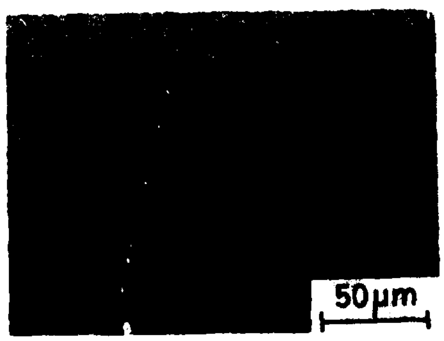

Fig. 20. Pitted roller surface of No. 3 bearing.

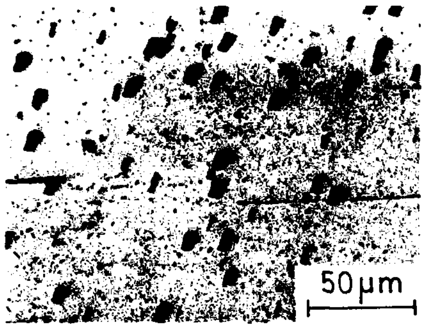

Fig. 21. Rectangular pits on the inner ring raceway of No. 3 bearing.

Fig. 22. Outer ring raceway surface of No. 3 bearing.

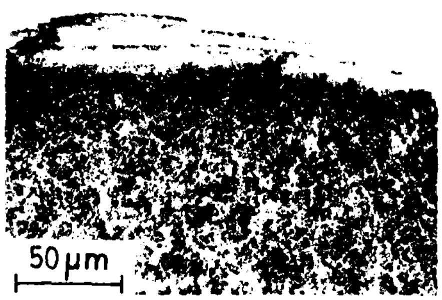

Fig. 23. Optical micrograph of different etching regions (LER and DER) in a roller of No. 3 bearing.

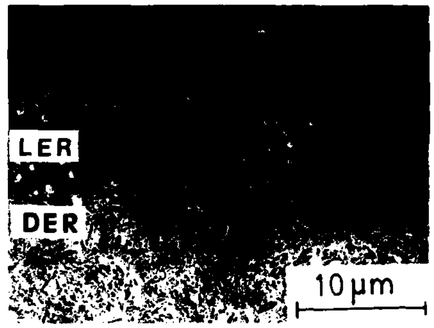

Fig. 24 SEM micrograph of LER and DER shown in Fig. 23.

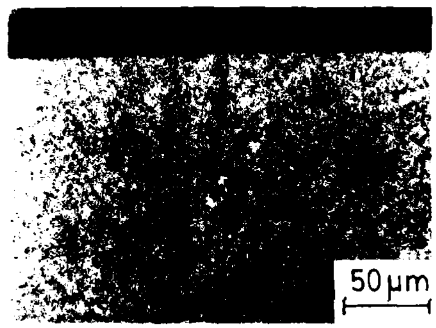

Fig. 25. A crack in a roller of No. 3 bearing.

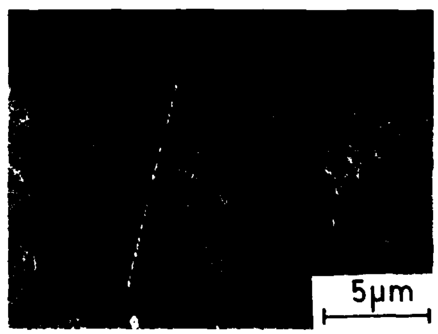

Fig. 26. Surface microstructure of a roller of No. 3 bearing.

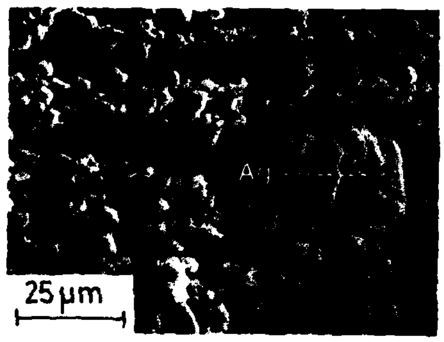

Fig. 27. Wear debris particles collected from No. 3 bearing at 350 hours.

MILITARY AIRCRAFT PROPULSION LUBRICANTS - CURRENT AND FUTURE TRENDS

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ABSTRACT

An assessment of the performance of MIL-L-7808J and MIL-L-23699C Military Specification lubricating oils in turbine engines and helicopter gear boxes is presented along with predicted performance of current and upgraded military specification oils in advanced and "growth" engine designs. Data is presented on advanced ester base engine lubricants, corrosion inhibited engine oils, and separate helicopter gear box oils evolving from current developmental research efforts. Future high temperature candidate fluids representing the ultimate stability for turbine engine oils are also discussed. Their use, in most cases, entails engine design considerations to accommodate their unique properties. The advantages and disadvantages of the various classes of synthetic lubricants for turbine engine applications are discussed, and deficiencies are identified where additional research programs are needed.

INTRODUCTION

Due to different environments and missions, the U.S. military services use different aircraft propulsion lubricating oils. For example, the U.S. Air Force has a low temperature operational requirement of -51°C (-60°F) while that of the U.S. Navy for gas turbine engine lubricants is -40°C (-40°F). Also, the U.S. Navy is generally more concerned with corrosion due to operating predominatly in a salt water ocean environment. Within the U.S. Department of Defense, the Air Force and the Navy have performed the development of lubricating oils for aircraft propulsion systems. This paper describes current aircraft turbine engine oils, several developmental turbine engine and helicopter oils, and anticipated future advanced oil development programs.

CURRENT OPERATIONAL ESTER BASED OILS

The present status of the lubricants used in U.S. military aviation gas turbine engines indicates that MIL-L-7808J (Ref. 1) and MIL-L-23699C (Ref. 2) oils are fulfilling service requirements. Visits to engine overhaul facilities generally reveal satisfactory cleanliness in lube system components and laboratory analysis of stressed oils obtained through service sampling on state-of-the-art aircraft indicate very low levels of lubricant degradation. The service discrepancy most reported is the chronic high rejection rate of mainshaft bearings due to corrosion. Based on these service reports the conclusion is that the current MIL-L-7808J and MIL-L-23699C ester based formulations are providing adequate protection against the thermal and oxidative degradation mechanisms existing in today's engines. The sole weakness in the present oils seems to be in their inability to thwart the static corrosion of bearings during long periods of engine inactivity. Although current MIL-L-23699C oils are expected to continue to be adequate in existing U.S. Naval aircraft, even with the normal engine improvement and "growth" programs which inevitably occur with most military engines, it is anticipated that certain future U.S. Air Force aircraft will require an advanced performance oil. This has led to the so called "4 cSt oil" developmental program which will be discussed later.

MILITARY SPECIFICATION UPGRADINGS

However, even while the current state-of-the-art engines are now entering service, the next generation of military gas turbine engines (circa 1990) is in development and these engines may not be so easy on the lubricant. Trends in aircraft gas turbine engine design show manufacturers taking advantage of material and technology improvements to build machines with higher pressure ratios and increased turbine temperatures in order to maximize fuel efficiency. Herculean efforts have been taken to obtain fractions of efficiency percentage point improvements by minimizing the amount of cycle air used for the cooling of bearing sumps and for seal buffering. These increased turbine temperatures and reduced cooling air flows translate into higher bearing compartment temperatures with the very real possibility of causing significant thermal and oxidative degradation of the lubricant including localized oil coking. In addition, improved bearing compartment sealing designs have reduced oil consumption to almost nothing. Since significant oil additions will no longer be required, the continual replenishment of the make-up oil (Ref. 3) will not occur and the antioxidant level will eventually be depleted. This improvement in oil consumption will resurrect an old, and, in this day of on-condition monitoring, an almost forgotten maintenance requirement, the scheduled oil change.

These next generation engines are being designed for use with typical MIL-L-7808J and MIL-L-23699C oils and, therefore, will be required to operate with any of the products now available. Since these specifications are performance specifications, i.e. they establish only certain minimum standards, it is reasonable to expect that there is a range of quality over the many products available. It can also be expected that engine/lubricant operation will reflect this range providing very good service with some oils and just acceptable results with others. Some products on the current Qualified Products List merely meet the published standards while others far exceed the expected level of quality.

MIL-L-23699C UPGRADING

Among the MIL-L-23699C oils are two "high quality" products recently developed primarily for use in the new high fuel efficiency engines being used in the commercial airline industry. Table I shows a comparison of the corrosion and oxidative stability and cleanliness characteristics of the two "high quality" products against the MIL-L-23699C specification and against average values for five typical qualified oils. It is apparent from Table I, particularly at the higher temperature oxidation tests, that improved quality MIL-L-23699C products are currently available. To insure that US Naval aviation gas turbine engines will continue to have the proper lubricants for the needed application, the US Navy will revise MIL-L-23699C to provide the improved cleanliness and thermal and oxidative stability needed for reliable operation in these next generation engine designs. While the specification revisions are still at least five years away, the anticipated cleanliness and thermal and oxidative stability requirements can be expected to be similar to those displayed by oils "A" and "B" of Table I.

MIL-L-7808J UPGRADING

The U.S. Air Force went through an upgrading process with the issuance of MIL-L-7808J in May 1982 whereby the minimum oxidative stability test duration requirement was doubled at 200°C (392°F) from 48 hours to 96 hours. This level of performance is expected to be adequate for U.S. Air Force aircraft for the next several years. However, it is anticipated that future aircraft engine systems such as the Joint Advanced Fighter Engine (JAFE), could benefit significantly by the development of an improved high temperature ester lubricant. This oil would also need to satisfy the U.S. Air Force world-wide operational low temperature extreme design criteria of -51°C (-60°F) defined by MIL-STD-210B (Ref. 4). In other words, the goal is to develop the highest temperature ester lubricant achievable which has -51°C (-60°F) pumpability. Thus an exploratory development program was initiated by the U.S. Air Force in 1984 to develop an aircraft turbine engine oil that would have better high temperature performance capability than current MIL-L-7808J ester based oils. This developmental engine oil will be referred to as the 4 cSt oil. Also described is an earlier program which led to the development of a MIL-L-27502 oil (Ref. 5).

MIL-L-27502 DEVELOPMENT

In the early 1970's, Air Force Materials Laboratory sponsored research at Monsanto Research Corporation and successfully developed a high temperature engine oil which through laboratory tests has shown potential capability for use over a

-40°C to 240°C (-40°F to 464°F) temperature range. However, its capability has only been demonstrated in an engine test at 200°C (428°F). Before its use at 240°C (464°F) can be endorsed, higher temperature engine validation testing would need to be conducted. This work has been previously unpublished except in U.S. Air Force technical reports (Ref. 6). This oil would have great improvement over MIL-L-7808 at the expense of some compromise in the low temperature performance. The specification values of MIL-L-27502 (slightly modified from the original fluid development program target requirements) are presented in Table II.

The selected candidate base oil was a blend of commercially available neopentyl polyol esters. It was selected based on three critical properties: 1) oxidation-corrosion resistance, 2) viscosity-temperature properties, and 3) storage stability. See Table III. Commercially available base stocks were screened for oxidation stability by formulating with an optimized additive package and subsequently evaluated in the corrosiveness and oxidation stability test. The 260°C (500°F) viscosity was set at 1.0 cSt minimum and the -40°C (-40°F) viscosity was set at 17,000 cSt maximum which ruled out many of the base stocks. Blending of lower viscosity esters with thicker esters, however, was also an approach used to increase ester viscosity, and was in fact used for the final selected candidate. Storage tests of formulated esters were also critical base oil screening tests.

Considerable effort under this contract was in selecting the right balance of additives. The final formulation which underwent turbine engine validation consisted of:

1. a neopentyl polyol ester blend
2. a deposit inhibitor (Ref. 7)
3. a heterocyclic amine oxidation inhibitor
4. dioctyldiphenyl amine, oxidation inhibitor
5. triphenylphosphine oxide, metal deactivator and synergistic antioxidant
6. dimethyl silicone, 350 cSt, antifoam additive

This formulation met the laboratory bench scale specification requirements as shown in Table II, with several exceptions which are small differences and are noted as follows: 1) low temperature viscosity: 17,643 cSt vs 17,000 cSt (15,000 cSt initially) maximum target goal at -40°C; 2) FS rubber compatibility: 4.2% swell vs 5 to 25% target range; and 3) foam test: sequence II foam volume 30ml vs 25 ml target foam volume. The original foam test performed at Monsanto met the requirement, but after transport to Wright-Patterson Air Force Base, the value of the second sequence was over the limit. In light of the excellent results, especially oxidation corrosion, bearing deposition and gear load carrying results, this candidate was tested (Ref. 8) by the Aero Propulsion Laboratory for 100 hours in a full-scale J57-P29W engine test conducted in accordance with MIL-L-27502.

The MIL-L-27502 engine test procedure is similar to that required by MIL-L-7808J except that the number 6 sump cover temperature is controlled at 300°C (572°F) and the bulk oil temperature is maintained at 220°C (428°F). Due to the high oil consumption attributable to the high bulk oil temperature, the oil normally lost through the overboard breather is collected and returned to the engine oil tank. The post test visual inspection of the completely disassembled engine indicated no evidence of corrosion or abnormal wear. Carbon deposits were rated medium which is considered relatively clean for such high operating temperatures.

Results of the 100 hour used oil analysis are presented in Table II. Overall the results are considered favorable. The largest change was in viscosity which increased 16% at 260°C (500°F) and 84% at -40°C (-40°F). Such a viscosity increase under the conditions of this engine test is not considered prohibitively excessive. The 100 hour used oil still met the new oil specification requirements of the corrosiveness and oxidation stability test at 220°C (428°F) and also at 240°C (464°F) except for bronze corrosion. Both the gear load carrying capacity and the bearing deposition test indicated very little difference between the 100 hour used oil and the new oil.

In summary, this 100 hour MIL-L-27502 engine test indicates that this oil formulation has excellent potential for high temperature turbine engine applications not requiring -51°C (-60°F) low temperature start up capability.

4 cSt OIL DEVELOPMENT

The target property requirements selected for this engine oil development program are shown in Table IV. The program objectives were believed attainable through a careful selection of the highest stability ester base stock combined with a

critical balance of performance improving additives. The basis for this belief was the successful development of the MIL-L-27502 engine oil and earlier ester studies performed by the Air Force Materials Laboratory. In light of the base oil and additive package proven for the MIL-L-27502 gas turbine oil, advancement to the target requirements shown in Table IV, was considered evolutionary in nature to the highest stability of an ester based oil possible while still meeting the -51°C (-60°F) low temperature performance criteria.

The viscosity-temperature requirements shown in Table IV reflect usability at the low temperature, less than 20,000 cSt at -51°C (-60°F), and adequate hydrodynamic film strength at the high temperature, greater than 4 cSt at 100°C (212°F). Figure 1 displays the approximate maximum transient bulk oil temperature range capability of currently used military specification turbine engine oils compared to that of the 4cSt oil. The other requirements in the Table IV reflect expected performance from an ester based fluid based on MIL-L-7808 and/or MIL-L- 27502 performance. The most difficult to achieve are the oxidation-corrosion test requirements and the deposit formation requirement, which are often related. The additives used must be effective in inhibiting oxidation, but must not promote deposit formation. It should be noted that the target properties are to an extent flexible and could be revised during the program if deemed necessary by the U.S. Air Force.

A letter was sent to industry requesting samples of base oils, additives and fully formulated fluids targeted to meet the requirements. Response has been highly encouraging. Material samples have been received from industry and many other companies are reportedly performing internal research from which we have not yet received samples. The comments from potential material suppliers has ranged from pessimistic i.e., the program goals are unattainable, to optimistic i.e., the program goals are challenging but attainable.

The ester base stock viscosity-temperature properties required to meet the target properties of the formulated product are achievable by appropriate ester blends. Such a base stock sample has been received from industry and properties are in Table V. Formulation with additives thickened the final formulation, as demonstrated by the preliminary data shown in Table IV on a formulation containing one of the more attractive additive packages. This formulation is continuing to be improved on a reiterative basis. Total target property compliance is believed to be highly probable or close enough to require only minor changes in the targets.

Based on this work, engine simulation evaluation is expected to begin in 1985 and actual engine testing is planned for 1986. Successful completion of these phases will then lead to transition for aircraft demonstration. Assuming successful progress, we expect to begin converting all MIL-L-7808J applications to the 4 cSt oil in 1988.

One of the advantages of this new oil is that it will be totally compatible and acceptable for use with all existing hardware now using MIL-L-7808 as well as the growth versions of these engines which will need or at least benefit from its improved high-temperature performance. Also when the 4 cSt oil becomes available with proven performance advantages, new engines can be designed to operate at higher temperatures for more efficient performance with less concern about hot spot coking and other oil degradation.

CORROSION INHIBITED TURBINE ENGINE OILS

While both the U.S. Navy and the U.S. Air Force have conducted research to develop corrosion inhibited turbine engine oils, there is a significant difference in their intended applications. The Navy program is directed toward the development of fully operational oils completely meeting MIL-L-23699C which also provide adequate corrosion protection throughout the drain life of the oil. The Air Force program is intended to provide corrosion protection in new MIL-L-7808 oil for use in cruise missile turbine engines for at least 30 months without engine operation. Then after storage, the oil must also function satisfactorily as a lubricant for a one time mission of a relatively short duration. In other words, the Navy program emphasizes the need for long term operational performance with corrosion protection whereas the Air Force program emphasizes the long term dormant corrosion protection followed by short term operational performance.

CORROSION INHIBITED MIL-L-23699

Current and next generation gas turbine engines using MIL-L-23699 lubricants are expected to share a common problem: static bearing corrosion. An on-going U.S. Navy program to develop a corrosion inhibited gas turbine engine oil has not been entirely successful. Candidates meeting the corrosion inhibited properties did not meet all

of the requirements of MIL-L-23699C, failing in one or two critical areas: load carrying capacity and/or compatibility. In all the oils examined the corrosion inhibited additive system had some adverse effect on the thermal-oxidative stability of the product. Since the MIL-L-23699C specification will be revised by 1990 to reflect the increased thermal-oxidative stability and cleanliness requirements needed for the next generation of engines, it seems unlikely that a suitable corrosion inhibited product will be developed which can meet these more strenuous limits. The present corrosion inhibited program is therefore being re-examined. Since the cost to replace bearings rejected due to corrosion remains very high, approximately three million dollars per year, efforts will continue to address a means to prevent such corrosion. Current ideas being considered are the possibility of using improved preservation maintenance techniques, i.e. dessicants, the use of corrosion resistant ion-implanted bearing materials and the re-introduction of preservative oils for limited flight use and for shipping.

CORROSION INHIBITED MIL-L-7808

A corrosion inhibited operational gas turbine engine oil was needed for the Air Launched Cruise Missile because of the unique application of the engine oil in this system. The missiles are required to operate satisfactorily after thirty months of storage. A storage oil is available, MIL-C-8188C (Ref. 9), but it is not an operational lubricant. It was designed to be drained and replaced with MIL-L-7808 at the time the system is to become operational. MIL-C-8188C contains an additive package for storage which causes the deposit forming tendencies, corrosion-oxidation properties and foaming characteristics to be unacceptable compared to current MIL-L-7808 operational fluid. The goal of this program was to develop an oil with corrosion protection equal to or better than MIL-C-8188C storage oil and with other properties equal to or better than those of MIL-L-7808H operational oils.

This program was Air Force sponsored at Pratt and Whitney Aircraft Group, Engineering Division and has been previously reported in the literature (Ref. 10, 11). The approach of the program was to develop an appropriate additive package for corrosion inhibition, blended into existing MIL-L-7808H engine oil. Over one hundred additives were screened both alone and in combinations with another additive. Initial screening of soluble additives consisted of anticorrosion protection, followed by acid number and flash point determinations. Many of these formulations exhibited excessive foaming characteristics, which was unacceptable. The sludge formation of candidates in the corrosion oxidation tests was another eliminating factor. A reiterative process was employed on marginal formulations.

A final candidate formulation was selected which contained 0.75% basic barium dinonylnaphthalene sulfonate and 0.25% alkenyl succinic acid as the corrosion preventive additive package. The properties of this fluid are presented in Table VI, compared to the MIL-L-7808H specification requirements. The corrosion protection of this candidate was equal to or better than that of MIL-C-8188C as determined by the Humidity Cabinet Test. While the total acid number of this candidate is 0.92 mg KOH/g, compared to the MIL-L-7808H requirement of 0.30 mg KOH/g, this was considered acceptable to continue with the more involved bearing deposition test. The post-test corrosion oxidation total acid number change of only +1.37 mg KOH/g, compared to the requirement of 4.0 mg KOH/g maximum, served to reassure that the original 0.92 mg KOH/g total acid number was not a major issue.

The bearing deposition test showed no adverse effects from the additive package. The deposit rating, viscosity change and acid number change were all equal to or less than the oil without the additive package. This was further demonstrated in a 100 hr J57 engine simulator test where the deposition and oil degradation characteristics of the candidate oil were again equal to or better than the oil without the corrosion inhibitor package. The only penalty attributable to the corrosion inhibitor additive package is a slight reduction (10%) in gear load carrying capacity. This is not considered disadvantageous since the gears and bearings in the intended Air Launch Cruise Missile engine application are not highly loaded.

NON-ESTER BASED ADVANCED OIL DEVELOPMENT

While ester based lubricants are satisfactory for the existing and next generation of engines, lubricant manufacturers indicate that the best of ester basestock and additive technology can only provide a modest improvement in the overall high temperature capability of this class of oil. Yet trends for the long term engine designs (circa 1995 and beyond) indicate that these engines will operate at significantly hotter internal temperatures in order to obtain the operational performance desired. The higher bearing compartment temperatures projected for these future engines will thermally stress ester based oils past their breaking point resulting in severely degraded oil and "dirty" compartments. It is, therefore,

apparent that in order to develop these engine designs improved non-ester based lubricants are required.

If, in the continued quest for improved performance in aerospace turbine engines, the operating temperatures of future engines continue to increase, as the trend appears to be, these temperatures will likely eventually exceed the maximum temperatures for liquid lubricants. Indeed, if we are limited to the ester based fluid technology, we are nearly to the maximum oxidative/thermal stability, as described in earlier parts of this paper. However, if we can consider significantly different chemical classes of basestocks, it is likely that the upper temperature limit of liquid lubricants can be extended by approximately 125°C (225°F) to the range of 350°C (662°F) to 370°C (698°F) bulk fluid operational temperature. The maximum operational temperatures as discussed in this section of the paper, refer to their maximum stability for extended periods of time in an oxidative environment. If future engines could be designed such that oxygen could be completely excluded from the lubricant, other chemical classes of fluids could be considered than will be discussed here. The temperature capability of the various classes of fluids to be discussed herein does not factor in the viscosity limitations as might influence load carrying ability. Because these fluids are so far away from realization as fully formulated candidate gas turbine engine oils, incorporation of factors other than low temperature viscosity and high temperature oxidative stability is not considered appropriate.

A non-ester based high temperature gas turbine engine oil was developed several years ago and its properties are described in Military Specification MIL-L-87100 (USAF) (Ref. 12). This lubricant is based on the polyphenylether class of fluids. This fluid is capable of use at temperatures up to 300°C (572°F), but has one major limitation, low temperature fluidity. The fluid as described in the military specification has a pour point of approximately +5°C (41°F) which represents a significant disadvantage if an engine using this lubricant were to be designed for world-wide deployment for which the extreme low temperature requirement for land based operations is -51°C (-60°F). Extensive attempts to improve the low temperature fluidity of the polyphenylethers both by formulation and by chemical modification of the molecular structure have been unsuccessful. While some improvement in the low temperature properties of the fluids may have been achieved, this improvement has not been achieved without significantly reducing their upper temperature thermal and oxidative stability. Therefore, unless some new, innovative way is found for improving the low temperature fluidity of the polyphenylethers without adversely affecting their upper temperature stability, they do not represent a very encouraging approach to the high temperature gas turbine engine lubricants required for the future.

The most promising chemical class of fluids for future high temperature gas turbine engine oils is the perfluoropolyalkylethers (PFAE). They possess inherent oxidative stability, thermal stability, good liquid range and they are non-flammable (Ref. 13, 14). Typical properties for both the branched and non-branched PFAE fluids are shown in Table VII. One of the early deficiencies that was found with these fluids was their tendency to be corrosive toward ferrous alloys at elevated temperatures in oxidative atmospheres. This tendency was reduced by the development of compatible, soluble additives which at very low concentrations (0.5-1.0%) stabilized the PFAE fluids by approximately 40°C (72°F) (Ref. 15). This stabilization is shown in Table VIII. As can be seen from the data, these fluids do show great promise for use at high temperatures. However, we should not be lulled into a false feeling of security that these fluids are nearly available and ready for use. There are still a significant number of factors that must be addressed and they are very basic problems. Many of the bench tests that are used in the assessment of a candidate fluid's potential as a gas turbine engine oil were developed using hydrocarbon based fluids and formulations. Based on our experience in a research program to develop a non-flammable hydraulic fluid, for which the primary candidate fluid is a chlorotrifluoroethylene (CTFE) based fluid, the chemistry of base fluids is not always adequately assessed in the standard tests (Ref. 16, 17, 18). For example, the lubricity of a CTFE formulation has been found to be superior to standard hydraulic fluids, MIL-H-5606 and MIL-H-83282, using the four-ball wear tests required by these military specifications. However, when this superior lubricity was assessed in standard aerospace hydraulic pumps, the hydrocarbon based fluids were found to be far superior. Another example found with the CTFE fluid, which is also totally halogenated like the PFAE fluids, was the need for a rust inhibitor which again was only found during component tests, although the standard stability tests including the presence of water would have been expected to reveal this potential problem based on our experience with hydrocarbon based hydraulic fluids. It is anticipated that similar deficiencies may be found with the PFAE based turbine engine lubricants as they progress from laboratory bench tests to component and hardware tests.

Another major difficulty when dealing with the PFAE fluids is their poor solvency for and response to conventional performance enhancing additives. It has

been our experience that when an additive is needed to improve some deficiency of the PFAE fluids, a research program is required to: 1) determine a class of additives that will provide the required improvement, and 2) synthesize a molecular structure that is soluble in the PFAE fluids. This is not meant to indicate that the task ahead to develop the PFAE fluids into high performance, high temperature gas turbine engine oils to meet the ever-increasing requirement imposed by future engines is impossible. But it is a significant challenge and the research should be initiated on a multi-disciplinary basis as soon as possible.

TRANSMISSION AND GEARBOX OIL DEVELOPMENT

Aside from use in aircraft gas turbine engines, MIL-L-23699C and in some instances MIL-L-7808J oils are also used in the gearboxes of helicopter power drive systems (e.g., input, main, intermediate, tail rotor and accessory gearboxes). In the early days of gas turbine powered helicopters the ester based synthetic oils worked fine in both the engine and gearbox systems. However, in today's helicopter transmissions the MIL-L-23699C and MIL-L-7808J engine oils are providing only marginal performance. Overhaul depots are reporting increasing rates of rejection of helicopter bearings and gears due to surface distress, corrosion and wear. In addition, the helicopter manufacturers are handicapped with the requirement to use military specification engine oils in new development programs which inhibits the gearbox design, reduces system durability and adds to aircraft weight. Adding to the frustrations encountered with the use of military specification oils are the field reports from commercial helicopter operators, using similar aircraft, who claim improved gearbox overhaul lives and lower maintenance actions resulting from the use of non-military specification oils.

The U.S. Navy has recognized these problems and has instituted a three phase program to improve helicopter transmission life and durability through the use of improved lubricants. The project phases are the 1) Interim, 2) Optimum and 3) Advanced Helicopter Transmission Oil Programs.

INTERIM OIL PROGRAM

The first phase of the project is to provide a helicopter transmission system oil with improved load carrying capacity to aid those gearboxes now experiencing marginal lubrication problems. This goal is being achieved by using existing commercial gas turbine engine oils with high load carrying capacity and years of successful aviation experience as the quickest means to introduce an effective and compatible oil into service. The Interim Oil is intended to be a transition fluid between MIL-L-23699 and an optimum helicopter transmission oil. It will provide a slight improvement in helicopter gearbox durability and, since the interim oil will not harm turbine engines if inadvertently mixed with the engine oil, it also will allow oil servicing personnel an interim period of time for training and adjustment to the concept of using a different oil in the gear box. This method of introducing a new fluid into operation should, therefore, be as smooth as is conceivably possible.

Preliminary copies of the Interim Oil specification were distributed to lubricant, engine and helicopter manufacturers in October 1984. The final version is now being prepared for publication. Two candidate products have passed all the requirements and will be listed on the Qualified Products List (QPL) of the specification when it is issued.

The primary differences between MIL-L-23699C and the Interim Oils are the increased Ryder gear rating, a modified silicone rubber compatibility test and the expanded viscosity change limit in both the corrosion and oxidation stability test at 205°C and in the Type 1-1/2 bearing rig tests. A comparison of these properties are given in Table IX.

OPTIMUM OIL PROGRAM

The second phase of the project will develop a separate lubricant specifically for use in current helicopter gearbox systems. It is this program which will give the maximum benefit to the helicopter community by providing an oil with high load carrying capacity and corrosion inhibiting properties to improve both gearbox durability and overall aircraft readiness while reducing costly part replacements due to corrosion and wear. The actual characteristics of the Optimum Oil are not yet defined, but many of the properties may be speculated upon. Since the oil is to be used as a gear lubricant certain high temperature properties needed for gas turbine engines can be reduced while those properties essential for durable gearbox operation can be optimized. Some of the materials and characteristics being considered are listed below:

a. Material Composition. The base fluid for the Optimum Oil has not been defined. Since the fluid will operate at modest bulk oil temperatures (typical current day designs have maximum limits of about 125°C (257°F)) thermal decomposition of the oil will not be a problem and the use of an ester-based fluid is not absolutely required. The use of a glycol or a synthetic hydrocarbon (polyalphaolefin (PAO)) based fluid has been suggested as a possible basestock material for this oil. The natural corrosion inhibiting properties and thermal-oxidative stability of the basestock material will be a large factor in selecting the most suitable fluid.

b. Additives. The fluid selected for the Optimum Oil will also need additive components to provide the load carrying capacity and the full amount of oxidation and corrosion inhibiting protection required for this lubricant. Current gas turbine lubricant additive systems use a proportionately large amount of antioxidants and metal deactivators to protect the oil from the severe thermal-oxidative environment. Experience gained in gas turbine oil development programs shows that attempts to improve the load carrying capacity and/or the corrosion resistance of these oils with current technology additives provides mixed results. Improved load carrying capacity or corrosion resistance are obtainable but only at the cost of degrading other essential characteristics (e.g. reduced thermal and oxidative stability, increased deposition, increased sediment (poor storage stability) etc.). In addition, many load carrying capacity additives severely attack elastomeric materials, particularly at high temperatures. Since the thermal environment for the Optimum Oil will be less severe than that of a gas turbine engine it can be expected that an entirely different additive package may be used. The conditions in current helicopter gearboxes are relatively mild compared to those in engines. Consequently, in the additive system of the Optimum Oil, the proportional amounts of antioxidants versus the amounts of load carrying capacity and corrosion inhibiting additives can be adjusted to provide the desired product improvements while still maintaining adeouate thermal and oxidative protection for the basestock fluid.

c. Properties. Quantitative properties of the Optimum Oil have not been established. However, by using MIL-L-23699C as a base fluid, some qualitative properties can be identified and are listed in Table X.

ADVANCED OIL PROGRAM

The final phase of the project is aimed at advanced transmission designs requiring high temperature stability with good load carrying capacity and corrosion inhibiting properties. The development of this class of helicopter transmission system is closely tied to concurrent advancements in lubricant chemistry and improved gear and housing materials which must operate at constant system temperatures of 260°C (500°F) and still provide good life. The success of such future helicopter designs will require the effort of several multi-disciplinary technologies acting together in a manner unlike that previously used for the design of conventional helicopter drive systems. Close cooperation between material engineers, lubricant developers and system designers is needed to insure the optimum success in such an undertaking. The technology needed for the production of such aircraft is still two decades away. However, communication between the industries involved needs to be started now if the project is to have any chance of success.

SUMMARY

The United States military gas turbine engine oil development efforts for current, near term future and long term future requirements have been discussed. The U.S. Air Force and U.S. Navy gas turbine engine oil operational environments are different enough to require several variations in the currently used formulated oils and in the anticipated future oils based both on esters and on more exotic fluids. These lubricating oils and related Navy transmission and gear box oil development programs have been reviewed and discussed.

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FIGURE 1

APPROXIMATE MAXIMUM TRANSIENT BULK OIL TEMPERATURE RANGE CAPABILITY FOR TURBINE ENGINES

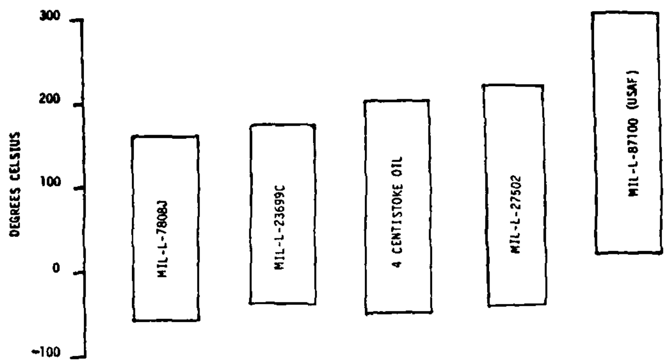

TABLE I

THERMAL AND OXIDATIVE STABILITY AND CLEANLINESS CHARACTERISTICS OF MIL-L-23699 OILS

Specification Test Item (selected parameters)	Spec. Limits	Typical (Avg. 5)	Oil A	Oil B
1. Corrosion & Oxidation Stability @				
a) 175 C				
-VIS change, %	-5/+15	+7.7	+1.7	+9.6
-TAN change, mg KOH/g	2.00	0.22	0.13	0.67
b) 203 C				
-VIS change, %	-5/+25	+21.2	+10.7	+14.2
-TAN change, mg KOH/g	3.00	1.67	0.90	0.89
c) 218 C				
-VIS change, %	Report	+80.1	+29.8	58.9
-TAN change, mg KOH/g	Report	14.34	6.56	10.27
2. High Temperature Bearing Rig Test				
-Deposit Rating	80 Max	44	4	12
-VIS change, %	-5/+30	20.3	16.0	19.0
-TAN change, mg KOH/g	2.00	1.20	1.30	0.91

TABLE II

MIL-L-27502 LABORATORY AND BENCH QUALIFICATION TEST RESULTS

SPECIFICATION TEST	REQUIREMENTS OF MIL-L-27502	NEW OIL	USED OIL DATA FROM 100 HOUR ENGINE TEST OF 0-77-20			
			25 Hrs	50 Hrs	75 Hrs	100 Hrs
Water Content - ppm	500 Max	4.2*				
Trace Sediment - ml/200 ml of oil	0.005 Max	.001				.001
Neutralization Number - mgKOH/gm	0.50 Max	.08				1.96
Specific Gravity - 15.6°C/15.6°C	Report	0.994*				
Viscosity at 260°C - cSt	1.0 Min	1.03				1.19
Viscosity at 98.9°C - cSt	Report	7.00				
Viscosity at 37.8°C - cSt	Report	40.1				52.6
Viscosity at -40°C - cSt 35 min	15,000 Max	17,643	21544	27264	36219	32910
3 hours	15,900 Max	--				
72 hours	17,000 Max	--				33240
Pour Point - °C	-54 Max	-54				-51
Shear Stability - % viscosity loss	4.0 Max	0*				
Flash Point - °C	246 Min	271				271
Autoignition Temp. - °C	410 Min	427				
Evaporation loss at 204°C - %	5.0 Max	1.3				
260°C - %	50 Max	15.8				
Specific heat at 60°C	0.40 Min	0.45*				
160°C	0.44 Min	0.53*				
260°C	0.48 Min	0.64*				
Foaming Characteristics - ml foam						
Sequence 1, 25°C - 5 min/60 sec	25/0	0/0* 10/0				10/0
Sequence 2, 93°C - 5 min/60 sec	25/0	15/0* 30/0				40/0
Sequence 3, 25°C - 5 min/60 sec	25/0	0/0* 0/0				10/0
NBR-H Rubber, swell - %	12 to 35	17.9				
F-A Rubber, swell - %	5 to 25	10.6				10.6
tensile strength - % chg	± 50	14				-13
elongation - % chg	± 50	7				19
hardness - chg	± 25	-5				5
FS Rubber, swell - %	5 to 25	2.3				1.6
tensile strength - % chg	± 50	-9				-4
elongation - % chg	± 50	-13				-9
hardness - chg	± 25	0				5
QVI Rubber, swell - %	No Req.	5.4*				

*Contractor Data

TABLE II (CONT'D)

LABORATORY AND BENCH QUALIFICATION TEST RESULTS

SPECIFICATION TEST	REQUIREMENTS OF MIL-L-27502	NEW OIL	USED OIL DATA FROM 100 HOUR ENGINE TEST OF 0-77-20			
			25 Hrs	50 Hrs	75 Hrs	100 Hrs
Corrosiveness and Oxidation Stability:						
48 Hours at 220°C (428°F)						
Viscosity Change at 37.8°C - %	25 Max	6.5				6.6
Neutralization Number Change	2.0 Max	0.8				-0.8
Metal Weight Change, Al - mg/cm^2	±.2	+.03				+.05
Ag	±.2	-.02				+.02
B_2 (AMS 4616)	±.4	-.04				+.02
Fe	±.2	-.07				+.08
M-50	±.2	-.06				+.10
Mg	±.2	-.05				+.07
Ti	±.2	-.05				+.02
48 Hours at 240°C (464°F)						
Viscosity Change at 37.8°C - %	100 Max	15.2				33.3
Neutralization Number Change	8.0 Max	4.4				6.15
Metal Weight Change, Al - mg/m^2	±0.2	-.06				+.02
Ag	±0.2	-.07				-.01
B_2 (CA 674)	±0.4	-.08				-2.65
Fe	±0.2	-.05				+0.05
M-50	±0.2	-.04				+0.01
WSP	±0.2	-.05				+0.02
Ti	±0.2	-.05				+0.02
Bearing Deposition Test - 240°C/300°C						
Avg. Demerit Rating/No. of Tests	80 Max	26/2				25
Filter Deposits Wt. - gms	2.5 Max	0.36				1.8
Oil Consumption - ml	3600 Max	1700				1800
Viscosity Change at 37.8°C - %	100 Max	30				45.5
Neutralization Number Change	2.0 Max	1.02				0.7
Metal Weight Change, Al - mg/cm^2	±0.2	-.1				0.0
Ag	±0.2	-.1				0.0
B_2 (CA 674)	±0.2	-.1				0.0
Fe	±0.2	-.1				0.0
M-50	±0.2	0.0				0.0
WSP	±0.2	-.1				0.0
Ti	±0.2	-.1				0.0

TABLE II (CONT'D)

LABORATORY AND BENCH QUALIFICATION TEST RESULTS

SPECIFICATION TEST	REQUIREMENTS OF MIL-L-27502	NEW OIL	USED OIL DATA FROM 100 HOUR ENGINE TEST OF C-77-20 25 Hrs	50 Hrs	75 Hrs	100 Hrs
LUBRICATION CHARACTERISTICS						
Gear Load Carrying Ability at 74°C	2400 Min	2825				2980
Gear Load Carrying Ability at 220°C	1000 Min	1009				

TABLE III

TARGET GOALS OF INITIAL SCREENING, MIL-L-27502 BASE OIL*

TEST	TARGET	
Corrosiveness and Oxidation Stability (96 Hours) at	220°C	240°C
Viscosity change at 37.8°C - %	15 Max	25 Max
Neutralization Number Change - mg KOH/g	2.0 Max	4.0 Max
Metal Weight Change - mg/cm²		
Al	±.2 Max	±.2 Max
Ag	±.2 Max	±.2 Max
Br**	±.4 Max	±.4 Max
Fe	±.2 Max	±.2 Max
M-50	±.2 Max	±.2 Max
Mg	±.2 Max	±.2 Max
Ti	±.2 Max	±.2 Max
Viscosity at 260°C - cSt	1.0 Min	
-40°C - cSt	17,000 Max	
Storage at 100°C - Days, No Precipitate	27 Min	
65°C - Days, No Precipitate	100 Min	

* Clark, F. S., Morris, G. J. and Reid, S. L, "New 465°F Turbine Oils," Unpublished Paper, 1976.

**Silicon Bronze (AMS 4616) at 220°C, Bronze Alloy (SAE-CA674) at 240°C

TABLE IV

TARGET AND CANDIDATE PROPERTIES FOR -51°C to 205°C
4 cSt GAS TURBINE ENGINE OIL

PROPERTY	TARGET REQUIREMENT	CANDIDATE	TEST METHOD
Kinematic Viscosity (cSt)			ASTM D 445
at 205°C	Report	--	
100°C	4.0 Min	3.96	
40°C	Report	17.14	
-51°C	20,000 Max	16,000	
Total Acid Number (mg KOH/g)	0.5 Max	0.39	ASTM D 664
Pour Point (°C)	-55 Max	-65	ASTM D 97
Flash Point (°C)	210 Min	255	ASTM D 92
Foaming Tendency (ml foam/ml foam after 60 second settling period)	100/0 Max	5/0	FTM 791b Method 3213
Autogeneous Ignition Temperature (°C)	350 Min	402	ASTM E 659
Evaporation Loss, %, 6.5 hr at 205°C	10 Max	3.1	ASTM D 972
Elastomer Compatibility, % Swell			ASTM D 3604
NBR -H	12-35	15.4	
FA	5-25	7.0	
FS	5-25	1.6	
QVI	5-30	13.0	
Vapor Pressure at 200°C (mm Hg)	10 Max	5.4	ASTM D 2879
Four Ball Wear Scar, mm			ASTM D 2266
52100, 75°C, 1 hr, 40 Kg Load, 600 rpm	0.7 Max	0.66	
M-50, 200°C, 1 hr, 40 Kg Load, 600 rpm	1.0 Max	0.51	
Deposit Forming Tendencies	0.5 Max	1.6	Fed. Test Method
Viscosity Change (%)	Report	124	Std No. 791b
Acid Number Increase	Report	8.34	Method 5003
Consumption, ml	Report	90	

TABLE IV (CONT'D)

TARGET AND CANDIDATE PROPERTIES FOR -51°C TO 205°C
4 cSt GAS TURBINE ENGINE OIL

PROPERTY	TARGET REQUIREMENT	CANDIDATE	TEST METHOD
Corrosiveness and Oxidation Stability 220°C, 48 hr,			FTM - 791b Method 5307.1
Viscosity Change (%)	25 Max	8.7	
Acid Number Increase	4.0 Max	1.13	
Metal Weight Change (mg/cm^2)			
Al	±0.2 Max	-0.1	
Ag	±0.2 Max	0.0	
Bz (AMS 4616)	±0.4 Max	+0.1	
Fe	±0.2 Max	0.0	
M-50	±0.2 Max	+0.1	
Mg	±0.4 Max	0.0	
Ti	±0.2 Max	0.0	
Shear Stability (% Viscosity loss)	4.0 Max	--	ASTM D 2603

PROPERTY	Goal	Max Accept.	Goal	Max Accept.	CANDIDATE	TEST METHOD
Bearing Deposition Test						
Deposit Rating	20	40	30	80	--	
Test Conditions Per MIL-L-	7808J		27502		--	MIL-L-7808J/27502
Neutralization Number Change	1.0 Max		2.0 Max		--	
Viscosity at 40°C, % Change	-5 to +15		-5 to +100		--	
Filter Deposits, g	1.0 Max		2.5 Max		--	
Oil Consumption, ml	1440 Max		3600 Max		--	
Aluminum Wt. Change, mg/cm^2	±0.2		±0.2		--	
Silver Wt. Change, mg/cm^2	±0.2		±0.2		--	
Bronze Wt. Change, mg/cm^2	±0.2		±0.2		--	
Iron Wt. Change, mg/cm^2	±0.2		±0.2		--	
M-50 Steel Wt. Change, mg/cm^2	±0.2		±0.2		--	
Waspaloy Wt. Change, mg/cm^2	±0.2		±0.2		--	
Titanium Wt. Change, mg/cm^2	±0.2		±0.2		--	

PROPERTY	Goal	Min Accept.	CANDIDATE	TEST METHOD
Gear Load Carrying Capacity				ASTM D-1947
Capacity, KN/m (ppi)	2550	2320	--	
Number of Determinations	4	4		

TABLE V

4 cSt ENGINE OIL BASE STOCK PROPERTIES

PROPERTY	CANDIDATE
Kinematic Viscosity - cSt	
at 100°C	3.83
40°C	15.81
-51°C	12,500
Total Acid Number - mg KOH/g	0.13
Pour Point - °C	-55
Flash Point - °C	232
Autoignition Temperature - °C	392
Evaporation Loss, 6.5 hr at 200°C - %	8.0

TABLE VI

COMPARISON OF MIL-L-7808H REQUIREMENTS AND BEST CANDIDATE CORROSION-INHIBITING FORMULATION

PROPERTY	MIL-L-7808H REQUIREMENTS	BEST CANDIDATE FORMULATION	TEST METHODS ASTM	TEST METHODS FED STD 791b
Kinematic Viscosity, cSt				
a. 98.9°C (210°F)	3.0 Min	3.54	D445	
b. -53.9°C (-65°F)			D2532	
@ 35 Minutes	17,000 Max	15,000		
3 Hour	17,000 Max	15,000		
72 Hour	17,000 Max	15,000		
Flash Point, °C (°F)	204 (400) Min	222	D92	
Neutralization Number (TAN)	0.30 Max	0.92	D664 (Modified)	
Foaming Characteristics				3213
a. Foam volume, ml	100 Max	15		
b. Foam collapse time, s	60 Max	5		
Evaporation loss @ 204°C (400°F), %	30 Max	10.4	D972	
Corrosiveness and Oxidation Stability @ 200°C (392°F) for 48 hours				5307.1
a. Change in Viscosity, %	-5 to 25 Max	+8.2	D445	
b. Change in TAN, mg KOH/g	4.0 Max	+1.37	D664 (Modified)	
c. Sludge, Volume %	Report	0.0		
Oil Deposit Rating	1.5 Max	0.2		5003.1
Bearing Deposition				
a. Overall deposit demerit rating	60 Max	34.6		
b. Change in Viscosity, %	25 Max	4.1	D445	
c. Change in TAN, mg KOH/g	25 Max	0.11	D664	
d. Filter Deposits, g	2.0 Max	0.49	(Modified)	
e. Oil Consumption, ml	1440 Max	400		

TABLE VI (CONT'D)

COMPARISON OF MIL-L-7808H REQUIREMENTS AND BEST CANDIDATE CORROSION-INHIBITING FORMULATION

PROPERTY	MIL-L-7808H REQUIREMENTS	BEST CANDIDATE FORMULATION	TEST METHODS ASTM	TEST METHODS FED STD 791b
Humidity Cabinet Test			D1748	
Hours till failure	Not Required	5 Panels 480 1 Panel = 370		
Engine (J57) Simulator Test, 100 Hrs				
a. Deposit Rating	Not Required	14.5		
b. Change in Viscosity, %	Not Required	10		
c. Change in TAN, mg KOH/g	Not Required	1.24		
Load Carrying Capacity			D1947	
a. Four Determinations, kN/m(lbf/in)	406 (2320)	370 (2110)		

TABLE VII

TYPICAL PROPERTIES OF BRANCHED AND NON-BRANCHED PFAE FLUIDS

FLUID	KINEMATIC VISCOSITY (cSt) -53.9°C -65°F	-40°C -40°F	37.8°C 100°F	98.9C 210°F	POUR POINT (°C)	(°F)	EVAPORATION, % WT. LOSS AFTER 6 1/2 HRS AT 204°C 400°F	260°C 500°F	288°C 550°F	316°C 600°F
LINEAR PFAE										
Fraction A	872	330	18	6.0	-54	(-65)				
Fraction B	7940	2875	132	42	-54	(-65)		0.32		55.6
Fraction C	24105	8675	376	113	-54	(-65)		0.32		100
BRANCHED PFAE										
Fraction AB	46000a	6900	85	0.2	-43	(-45)	5.0	27		
Fraction AC	b	42000c	280	25	-34	(-30)			12	34.8

a - at -18°C (0°F)

b - too viscous to measure

c - at -32°C (-25°F)

TABLE VIII

CORROSION AND OXIDATION STABILITY OF BRANCHED AND NON-BRANCHED PFAE UNFORMULATED AND FORMULATED FLUIDS

Temperature °C (F)	% Visc Change at 37.8°C (100F)	Fluid Loss Wt%	Weight Change (mg/cm^2) 4140	52100	410	M-50	440C	Formulation
Unbranched PFAE								
288 (550)	a	84	0.02	+0.48	5.57	-2.37	-3.10	None
288 (550)	+0.22	0.31	+0.04	+0.03	+0.05	+0.01	0.00	1% P-3
316 (600)	+0.10	0.25	+1.43	+0.41	-0.35	+0.44	-0.02	1% P-3
Branched PFAE								
316 (600)	+3.4	5.2	+3.11	+1.17	+0.72	+1.80	+0.46	None
316 (600)	+3.0	0.14	+0.13	+0.01	+0.01	+0.10	0.00	1% P-3
329 (625)	+4.8	0.22	+0.13	0.00	-0.02	+0.07	0.00	1% P-3
343 (650)	+2.3	0.50	+0.05	+0.12	+0.01	+0.31	+0.06	1% P-3

a - Insufficient Sample to Determine

TABLE IX

A COMPARISON OF CHANGED PARAMETERS BETWEEN MIL-L-23699 AND THOSE OF THE INTERIM HELICOPTER OIL

Parameter	MIL-L-23699	Interim Oil
1. Ryder Gear Test		
Relative Rating, % Hercolube A	102	152
2. Silicone Rubber Compatibility		
Test Temperature, C	121	110
Duration, Hours	96	96
Swell, %	+5 to +25	-5 to +25
Tensile Strength Loss, %	30 Max	60 Max
3. Corrosion and Oxidation Stability at 205°C		
Viscosity change @ 38°C, %	-5 to +25	0 to +30
Total Acid No. Change, mg KOH/g	3.0 Max	3.0 Max
Metal Weight Change,		
Steel	+/- 0.20	+/- 0.20
Silver	+/- 0.20	+/- 0.20
Aluminum	+/- 0.20	+/- 0.20
Magnesium	+/- 0.20	+/- 0.20
Copper	+/- 0.40	+/- 0.40
4. Bearing Test - Type 1-1/2		
Overall Deposit Rating	80 Max	80 Max
Viscosity Change @ 38 C, %	-5 to +30	0 to +35
Total Acid Number Change, mg KOH/g	2.0 Max	2.0 Max
Filter Deposits, g	3 Max	3 Max
Total Oil Consumption, ml	2000 Max	2000 Max

TABLE X

COMPARATIVE OPTIMUM HELICOPTER OIL PROPERTY CONSIDERATIONS

Property/Requirement	MIL-L-23699	Optimum Oil
1. Basestock Material	baseline ester	ester glycol synthetic hydrocarbon
2. Thermal and Oxidative Stability °C (°F)	baseline 175 (347)	reduced 125 (257)
3. Corrosion Inhibition	baseline	improved
4. Load Carrying Capacity (Ryder Gear Rating)	baseline	increased 2 X
5. Viscosity, 10 mm^2/sec (cSt) at	baseline	increased
99°C (210°F)	5.0 to 5.5	7.5 to 12.0
-40°C (-40°F)	13,000	20,000
6. Pour Point °C (°F)	baseline -54 (-65)	unchanged -54 (-65)
7. Foaming	baseline	unchanged
8. Sediment	baseline	unchanged
9. High Temperature Deposition, Type 1-1/2 Bearing Rig Test	baseline	not required

AD-P005 066

FUTURE TRENDS IN HELICOPTER TRANSMISSION LUBRICANTS

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SUMMARY

Some recent fundamental studies relating to the lubrication of helicopter transmissions are presented and their implications for future oil development discussed.

The resulting enhanced understanding of lubrication mechanisms has shown that real potential exists both for improved performance and for the ability to cater for higher temperatures, by means of the formulation and use of improved transmission oils.

If such benefits are recognised and if the new knowledge is correctly applied, there is no reason why helicopter gearbox reliability should not be appreciably improved by the end of the decade.

INTRODUCTION

A wide range of different lubricants is currently in use in helicopter transmissions. Table 1 shows the lubricants employed at present in the transmissions of just one, UK based, helicopter manufacturer (1). Fluids used range from mineral to synthetic based oils and from low to high viscosity.

This diversity reflects not so much differences in helicopter transmissions in use, but rather different solutions to the problem of protecting gears and bearings. Thus some lubricants are high viscosity, extreme pressure, mineral-based gear oils. Here the industrial gear oil solution has been adopted, of attempting to maintain as thick an elastohydrodynamic (EHD) film as possible and having anti-seizure additives to cope with EHD film breakdown. Other lubricants are lower viscosity, anti-wear gas turbine synthetics. These provide thinner EHD films and survive by powerful anti-wear protection.

It should not be imagined that the helicopter user has adopted these different solutions knowingly. As can be seen form Table 1, lubricants in current use were all designed for another purpose and later adopted for helicopter transmissions with, at most, minor modifications. Each lubricant has a particular mode of protection, developed systematically for the original application but now being applied to the different, and particularly demanding, role of a helicopter transmission oil.

As will be described in the next section, this ad-hoc approach to helicopter transmission lubricant selection has not proved particularly successful in the past. One of the most significant trends in helicopter lubricants currently taking place is the development of lubricants specifically for their intended role. Such work has taken place in the UK (2) and is presently being extended, and has also recently been started in the US.

If such lubricant development work is to provide effective solutions to the increasing demands of helicopter transmissions, however, it must, in the author's opinion, be based not just upon a great deal of test work but also upon a clear understanding of the mechanisms of lubricant protection and failure. The main aim of this paper is to describe fundamental, recent work which clarifies some of these mechanisms and to discuss how the resultant insights could shape helicopter transmission lubricants in the future. As well as looking at completed work, the author has also attempted to point out areas of ignorance where work remains to be completed.

REQUIREMENTS AND LIMITATIONS OF HELICOPTER TRANSMISSION LUBRICANTS

Helicopter transmissions possess a number of characteristics that make them particularly difficult to lubricate successfully.

Stringent weight control means that power/mass ratios are extremely high, compared to marine and industrial transmissions. At the same time, helicopter structures are flexible and subject to vibration. Thus localised tooth and bearing stresses are severe, variable and difficult to predict.

Helicopter transmissions generally have large speed reductions, from typically 20000 to 300 rpm, although sometimes first stage reduction takes place in the engine. This means that high and low speed gear and bearing systems, both transmitting high power, have to be lubricated by the same oil. Helicopter lubricants also require a wide effective temperature range. Typical operating temperatures are 70-90°C, but in warm climates or under prolonged high power use, temperatures up to 135°C may be reached. However helicopters should also be able to function at very low ambient temperatures and a pour point below -40°F is often stipulated for the lubricant.

A further constraint often imposed by military operators is that the helicopter gas turbine engine and transmission have a common oil, in practice the engine oil. The great limitations imposed by this requirement have been discussed previously by the author (1).

In view of the above conditions it is not altogether surprising to find that helicopter transmission oils do not usually operate as effectively as most other lubricants. Helicopter transmissions generally have low times between overhaul (TBO's) and even shorter mean times between removal (3). Life is often limited by either bearing failure in gear pitting. Micropitting is frequently found during overhaul and sometimes scuffing. However scuffing more often occurs when other transmission problems arise or during new gearbox development.

All the above problems indicate that current helicopter transmissions are operating at the limits of their capabilities and give only mediocre performance in terms of life. This is probably an inevitable result of the severe conditions in helicopter transmissions coupled with the haphazard way that their lubricants have been developed. Before improved lubricants can be designed, however, solutions must be found to the problems of coping with the wide temperature range, large reduction ratio and very high power densities characteristic of helicopter transmissions.

MECHANISMS OF DAMAGE IN HELICOPTER TRANSMISSIONS

Pitting

Pitting is found in both gears and bearings in helicopter transmissions. It is almost always surface initiated in bearings and generally so in gears.

One obvious influence that the lubricant can have on pitting life is through its effect in elastohydrodynamic film thickness and thence specific film thickness or "λ-value". Fatigue life has been shown to fall off rapidly at λ-values below 2.5 (4). With values lower than this, asperity interactions become progressively more numerous and severe (5) and this is generally presumed to help initiate fatigue cracks. Another mechanism for how low λ-values might accelerate fatigue may involve wear debris. It has been shown by Macpherson and co-workers (6), using a bearing test machine, that fine filtration causes a large increase in fatigue life. However little further benefit was gained by filtering below 2μm and this was ascribed to the fact that smaller particles were able to pass through an EHD contact without damaging the surfaces and thence initiate fatigue. Clearly such an effect would be λ-value dependent.

Lubricants have an influence on fatigue over and above that due to EHD film thickness, however. Different lubricant additives have been shown to have different effects on fatigue life (7), (8), though in a somewhat unpredictable manner. The influence of chemistry on fatigue life is well shown in a study by Macpherson (9), who looked at the response of of fatigue life to temperature in a disc machine. Using a formulated oil, containing a phosphorus anti-wear additive, pitting life was longer for tests at 80°C than for tests at 50°C inlet temperature. This is clearly contrary to theoeretical λ-value variations and is thus a chemical effect. A lubricant without additives gave no such rise in fatigue life (10).

A further way in which lubricant chemistry can affect pitting is by the presence of dissolved water. Small amounts of water can greatly reduce fatigue life. The amount of water which will dissolve in a lubricant varies greatly from base stock to base stock. What is not yet know is whether fatigue life reduction depends upon the actual water concentration in a given base stock, or upon the chemical activity of the water in that base stock. In the latter case, lubricants that naturally dissolve more water would not, in consequence, give lower fatigue lives than other lubricants. Work in this area is currently being carried out.

Scuffing

Although scuffing is not the prime mode of failure in helicopter transmissions in operation, the evidence suggests that such transmissions are operating close to their scuffing limit at present and that any improvement in pitting resistance that leads to higher contact pressures could expose scuffing as a major limitation. Scuffing is often a problem in overload tests during gear development, testing and running-in. It can also occur between the rib and roller ends in taper roller bearings.

Until quite recently most scuffing models were boxed around the critical temperature hypothesis of Blok (11) or a related criterion such as critical power intensity (12). This type of model has been applied with some success to the prediction of scuffing in straight oils (13), but has not been effective for lubricants containing anti-wear or ep additives.

In the last few years a new group of models, applying EHD to scuffing, have been developed (14), (15). These regard scuffing as a sudden breakdown in the effectiveness of the EHD component of a mixed EHD-boundary film. At some point, it is suggested, due to a combination of thermal effects and to the load being supported by solid interactions rather than fluid pressure, the lubricant can no longer reach its normally high viscosity in the inlet zone, and the fluid component of the load support collapses.

These EHD - based models are not yet predictive tools, a major problem being that they contain surface roughness as a component. This introduces a constantly changing statistical feature, making it difficult to develop useful design equations.

Like the critical temperature model, EHD approaches to scuffing have difficulty in encompassing chemical effects due to lubricant additives. Currently, the proponents of EHD scuffing models tend to consider that lubricant chemistry influences scuffing primarily by helping to smooth the metal surfaces.

Micropitting

Recent work has provided a convincing mechanism of micropitting (16), and shown that it may not only be a possible precursor of fatigue pitting but may alternatively be followed by high rates of wear. Micropitting is a fatigue phenomenon on an asperity scale. It has been propsed that as opposing surfaces contact with a combination of normal and tractive force, plastic deformation occurs. After unloading, this results in residual stresses in the deformed region, which are tensile, i.e. crack opening. The model suggests that cyclic strain due to repeated asperity contact, coupled with these tensile stresses, promote the growth of micropits.

This model explains well the experimental observation that, on the driven gear, micropits slope shallowly into the surface away from the pitch line but on the driver they slope towards the pitch line (figure 1). These directions are normal to the calculated tensile stresses described above.

Since micropitting requires widespread asperity interactions, its occurrence depends upon there being a low λ-value.

Wear

Wear is not generally considered a problem in helicopter transmissions in operation. It is usually believed that gears can tolerate a good deal of wear before this causes other problems to arise. However tooth profiles are not often measured even on gears prematurely removed from service, so it is not possible completely to exonerate wear as a cause of failure.

There have been a number of reported cases of very high wear in gear and disc machine test work. This appears particularly prevelant with anti-wear oils with low viscosity, such as those conforming to MIL-L-23699B. A feature of this wear is that it takes place with hardened steels at very low slide/roll ratios. The mechanism of this type of wear has recently been studied and has been shown to be a fatigue process, relating to and consequent upon the micropitting mechanism described previously (17). In disc machine tests it has been shown that for rapid wear of a surface that surface needs to be slightly softer than the other and then, at low λ-values, wear rates of typically 10μm/hour can follow micropitting on the softer surface. This mode of wear appears to involve the extrusive removal of the weakly supported metal from above the shallow cracks described above. As this material is removed, the remnant of the crack tip remains and propagates further into the surface. The very high microcrack density ensures almost even, steady wear over the whole surface. Interestingly this wear rate does not increase significantly with increased slide/roll ratio and thus as sliding increases it tends to be pre-empted by pitting failure.

Fretting

Fretting occurs in helicopter splines and also in clamped assemblies commonly found in helicopter transmissions. In such assemblies the resultant minor dimensional loss can lead to gross wear and complete loss of torque on the bolts, with consequent failure.

Fretting involves oxidation of rubbing surfaces or debris and studies suggest that the lubricant can play a large part in its control. Well chosen antioxidants can reduce fretting, as can a lowering of the concentration of dissolved oxygen in the lubricant. It has been reported that fretting in a spline tester was virtually eliminated by correct oil formulation (2).

INFLUENCE OF LUBRICANTS ON SPECIFIC FILM THICKNESS

From the previous section it is clear that a recurrent theme is the importance of specific film thickness. A very effective way of controlling damage and failure is to increase λ-value.

It is difficult to calculate λ-values at all accurately in helicopter transmissions due to misalignment, vibration and general uncertainty as to surface temperatures. However estimates generally lie in the λ = 0.5 to 2 range. This is below the limit of about 2.5 where asperity contact becomes significant so that helicopter transmissions are very much in the mixed boundary-EHD regime, where increases in λ are especially beneficial.

The lubricant can affect λ values in several ways. These are now briefly discussed.

Viscosity

According to EHD theory (20), film thickness h_0 is related to inlet viscosity, η_0, by

$$h_0 \propto \eta_0^{0.7}$$

Thus a simplistic approach would suggest the use of the most viscous possible lubricant at the operating temperature in order to obtain high h_0 and thence high λ-value. With mineral oil this means a viscosity of about 15cS at 210°F. Higher viscosities are really not practicable since the relatively low viscosity indices of mineral oils would lead to unacceptably viscous fluids at low temperatures. With the DTD 900 oil in Table 1, a change to a lighter oil in winter is necessary. The higher VIs and lower pour points of polyglycols and synthetic hydrocarbons, however, make the use of a more viscous oil possible.

Unfortunately there are problems in using a very viscous lubricant, in a high speed transmission. Firstly shear heating, both bulk and localised within the EHD contact, can occur in a viscous oil and result in a nominal increase in oil viscosity providing negligible increase or even a decrease in EHD

film thickness (20). High viscosity oils also tend to be more susceptible to shear thinning at the very high strain rates prevalent in EHD contacts and this too can reduce film thickness.

Other problems can arise in high speed bearings. A viscous lubricant can often produce skidding of rolling elements, especially in relatively lightly loaded bearings. Studies of rolling element bearings have also shown that starvation is a major problem, especially using high viscosity oils. It probably has an even more deleterious effect on film thickness than heating (21), (22).

Thus a simple increase in viscosity of lubricant, although a possible solution in a simple gearbox, is not necessarily a satisfactory solution in a full helicopter transmission system. Other ways need to be found of increasing λ-value.

Pressure Viscosity Coefficient

A second possible method of increasing EHD film thickness and thence λ-value is to use a lubricant with a high pressure viscosity coefficient, α. According to EHD theory, film thickness is related to α by approximately

$$h_o \propto \alpha^{0.5}$$

The pressure viscosity coefficient is determined by base oil rather than additives and can vary considerably between base oils, as shown in Table 2.

The problem with this approach is that fluids with high pressure viscosity coefficients i.e. which rapidly become viscous when pressurized, also have high temperature-viscosity coefficients and thin rapidly when heated. This correlation can be seen in Table 2, where temperature-viscosity coefficient is expressed as ratio of viscosity at 38°C and 99°C. There are fundamental reasons why the responses of a fluid to pressure and temperature go hand in hand (23). Unfortunately helicopters lubricants require a low temperature-viscosity coefficient to cover the wide range of temperatures encountered in service. Thus pressure-viscosity coefficient needs have to be sacrificed.

Pressure viscosity coefficient also drops with rise in temperature and this must be taken into account when calculating film thickness. The overall effect of temperature on the ability of a lubricant to generate EHD films is conveniently expressed by a plot of "lubricant parameter", the product of viscosity and pressure-viscosity coefficient, against temperature. EHD film thickness can be taken as approximately proportional to this product to the power 0.7 (24).

Figure 2 shows lubricant parameters for two base oils with the same viscosity at room temperature, one with high and one with low pressure viscosity coefficient. It can be seen that the benefits of high pressure-viscosity are overwhelmed by the related high temperature-viscosity as temperature increases (25).

Thick Chemical Films

Recent work has indicated that film thickness in EHD contacts can be greatly increased above its theoretical EHD value by the formation of a chemical film by anti-wear additives on rolling/rubbing tracks.

Until a decade ago it was generally considered that anti-wear additives formed very thin layers on metals, a few atoms deep, and that their main influence on wear was from friction reduction and possibly smoothing. Recent studies of zinc dialkyldithiophosphates (26)(27) and phosphonate esters (28) (29), two of the most common types of anti-wear additive, suggest a different mode of action in which viscous iron-phosphorus ester polymer films up to 1μm thick form on rubbed surfaces.

With zinc dialkyldithiophosphates this film was just visible due to interference colours and included iron oxide (26). With phosphonate esters the film, having the same refractive index as oil, could not be seen but its presence could be inferred by optical interference film thickness measurements. Thus a separation remained even between static surfaces after a test, due to the presence of an invisible surface coating (28). This lack of visibility might explain why this mode of action of phosphorous anti wear additives has not been previously emphasised. There is evidence that it occurs in practical cases. Macpherson (30) in the fatigue study cited earlier, where performance increased at high temperatures noted, at high temperatures, "a thin but hard, varnish like layer on the discs", using formulated oils containing a phosphorous additive.

This type of behaviour has considerable implications for helicopter transmissions in that it could increase λ-value without using very viscous oils - with equal validity for high and low speed end of the transmission, and being particularly effective at high temperatures.

Chemical Smoothing

λ-value can be increased by reducing surface roughness as an alternative to increasing film thickness, h_o. This is complicated by the fact that roughness tends, on its own account to produce changes in EHD film thickness, depending upon how asperities are aligned relative to the direction of sliding (31) . Gear and disc surfaces are ground to typically 0.4μm CLA and smooth to at least 0.2μm CLA during running-in. Much of this initial smoothing is thought to result from plastic deformation but it is often observed that some lubricants produce smoother steady state surfaces than others (32).

Such lubricant-dependent smoothing, has not been systematically studied though some authors ascribe the scuffing resistance of additives almost entirely to their smoothing effects (15). It is not yet known what degrees of smoothing can be achieved in practical gear systems, nor, indeed, what degree is

desirable. A study at slow speeds has suggested that ultra-smooth surfaces can increase friction (34), but it is not clear whether this is relevant to mixed lubrication conditions. Very smooth surfaces are also deleterious in piston liners, because they prevent oil retention in the surfaces, but it is difficult to see how this is directly applied to splash lubricated EHD.

This effect of chemical smoothing has considerable potential but, of all the fields surveyed in the paper, needs the most research to be done before it can be applied as a deliberate tool in failure prevention, and helicopter gear oil development.

INFLUENCE OF THE LUBRICANT ON FRICTION

Another factor that recurs in mechanism of damage and which is likely to be affected by lubricant composition is coefficient of friction, in particular friction at asperity contacts. Friction is important in most models of scuffing and wear, and also in micropitting mechanisms. Its influence on pitting is less obvious, but small amounts of sliding are found greatly to reduce pitting lives and friction is likely to be implicated in this effect.

In mixed EHD contact, overall friction coefficient comes from a combination of asperity friction and fluid shear friction. Fluid friction, or traction coefficient, is typically in the range 0.03-0.08, depending upon the slide-roll ratio (figure 3). This value can easily be measured using very smooth surfaces so as to maintain an almost full fluid film. Fluid traction coefficient in EHD varies with fluid structure, as shown in Table 3. Coefficient of friction at contacting asperities in mixed EHD is much more difficult to determine. Some authors suggest a value of 0.1-0.2 for analytical work (35) (36). One experimental approach to the problem is to use very low speed friction tests, so that all the load is borne by asperities and to study the effect of lubricant composition on friction coefficient over a range of temperatures. This approach suggests that boundary films at low temperatures give friction coefficients of 0.08 - 0.15. Phosphorus anti-wear additives at intermediate temperatures give values of 0.12-0.15 and sulphur e.p. additives at high temperatures produce similar values. Using this technique, considerable differences in friction coefficient are found for different lubricants, suggesting that asperity friction may be improved by appropriate lubricant formulation.

Unfortunately, such slow speed friction coefficient values do not necessarily translate directly to realistic mixed-EHD conditions. Average load on asperities is likely to be higher in a slow speed test than a high speed gear, but power input may be considerably less. Also, slow speed tests allow one surface a much larger "out of contact time" to recover between successive contacts with the other surface. This might give chemical processes time to take place in slow speed tests which would not be possible at high speeds.

An attempt was made a few years ago to measure friction rises due to asperity contact events in a mixed EHD contact (37). A very viscous fluid at very slow rolling speeds was employed, to stretch the likely contact time of asperities up to the response time of a strain transducer. Little success was achieved, though the reason is not clear.

One possible explanation is that under normal EHD conditions true solid-solid contact is very rare, and instead micro-EHD films form between approaching asperities (38). Such films, if they exist must however, be so thin as to offer negligible electrical contact resistance since this is found in practice.

Thus although it appears likely that friction could play an important part in the various types of damage that can occur in gears and bearings, it has not yet been clearly demonstrated, as it has for λ-value, that lubricant chemistry plays an important part in reducing the friction under realistic EHD conditions. This is, however, most likely.

GENERAL DISCUSSION

The foregoing has shown that the combination of very high and very low speeds found in a helicopter transmission poses particular problems that are difficult to meet by relying primarily upon the viscosity of the lubricant. A thin oil does not provide sufficient film thickness in the flexible, high power, slow speed end. However a very viscous fluid is unsuitable at very high speeds. It has been shown, however, that there are a number of chemical effects which can help control damage and failure in mixed-EHD systems which are applicable equally to both high and slow speed transmissions. Some of these effects, such as chemical thick film formation, are reasonably well characterised. Others, such as smoothing and asperity contact friction need a good deal of further work.

The fact that such mechanisms exist has implications for the design of helicopter transmission oils. One is on the type of scuffing, or load-supporting test, employed in helicopter lubricant development work. The chemical processes outlined all require elevated temperature, a matching of lubricant type with steel type and time to occur. They are not extreme pressure (ep) behaviour in the classic sense as measured in a scuffing test rig. These rigs, such as the 4-ball ep tester, run for short periods under severe conditions using new specimens of an inappropriate steel. Anti-wear tests, such as the 4-ball wear procedure, are better, in that they permit time for chemical activity to occur, but such tests do not explore the subsequent load carrying capacity of the film formed and thus often tend to measure the rate of formulation of chemical protection rather than its eventual effectiveness. Tests for development of helicopter transmission fluids need to have the following.

- Intended steel for use
- Temperatures encountered in practice
- Extended periods of running prior to overload
- Out of contact times appropriate to practice.

Such conditions can be provided by disc machines, but also by simpler rigs if appropriately designed (2).

A second feature of the types of protection described earlier is that they tend to be applicable to the control of a range of types of damage. Thus increasing λ-value is beneficial in controlling both scuffing and pitting. There may be exceptions to this, where specific additives are particularly implicated in one type of failure, by corrosion, for example, but this general correlation between good pitting and good scuffing resistance has been noted by the author in gear oil development work (2). It suggests that such development work may concentrate in the early stages, when large number of candidates are screened, on the easier to test scuffing resistance and postpone time-consuming pitting studies until later in the program.

CONCLUSIONS

This paper has examined recent and current research in lubrication which the author considers relevant to the development of helicopter gear lubricants in the future. The author believes that the insights that this work provides could, if applied systematically, yield improved helicopter lubricants, with increased TBO's even for the exacting requirements of the next generation of helicopters.

In the past lubricants have not been developed with much attention to fundamental mechanisms of failure and protection. For all applications, except possibly those in space, lubricants have evolved based on past experience and routine test work. For a new application the question usually asked by an oil company of itself is - what type of lubricant can be modified to suit the situation? So far as helicopters are concerned the result has tended to be mediocre and helicopter users and manufacturers have grown accustomed to living with lubrication problems and the expenses consequent upon them.

It is the author's contention that the next few years may show that this state of affairs is not inevitable. Sufficient understanding of lubrication mechanisms has been acquired to improve on the "test and modify" approach to helicopter transmission lubricant development, especially with regard to coping with the differences in speed throughout a helicopter transmission.

In the ideal scenario the future will see such ideas being applied carefully and systematically to the design of better lubricants for helicopters or, even a single, effective lubricant to cater for all helicopters throughout the Western world. Such an outcome depends, however, upon helicopter manufacturers no longer being prepared to accept second best but to take steps to ensure that their requirements are properly met. Judging by past performance this may be wishful thinking.

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YEAR INTRODUCED	HELICOPTER	OIL TYPE	OIL SPEC.	VISC cS at 100°C	
58	Whirlwind	Mineral	D.ENG.RD.2479/1	9.1	Very early mineral gas turbine
59	Wessex	Ester	D.ENG.RD.2487	7.7	Gas turbine oil (Type 1)
62	Scout	Mineral	DTD 900/4981	17.5	Modified back axle oil
64	Sea King	Ester	MIL-L-23699B	5.25	Gas turbine oil (Type 2)
67	Gazelle	Mineral	DTD 581C	8.2	Aircraft oil
67	Puma	Mineral	DTD 581C	8.2	Aircraft oil
67	Lynx	Mineral	DTD 900/4981	17.5	Modified back axle oil
80	W.30	Mineral	DTD 900/4981	17.5	Modified back axle oil

Table 1 Typical Main Gearbox Lubricants in Current Use

	Pressure Coefficient (measured in EHD)	Temperature Coefficient $\nu_{38°C}/\nu_{99°C}$	Ref.
Dimethylsilicone* (1000cS)	1.1	2.5	37
Dimethylsilicone* (100cS)	6.4	2.44	37
Polyolester (Pentaerythritol tetravalerate)	7.5	4.55	38
Diester (di-2-ethylhexyl adipate)	7.6	3.45	38
Polypropylene glycol (UCON LB165)	14	5.26	38
Phosphate ester (2-ethylhexyl diphenylphosphate)	14	4.17	38
1,1,7-trihydroperfluoroheptyl-perfluoroglutarate	16	6.25	38
Mineral Oil (Paraffinic-Naphthenic)	22	11.5	37
Mineral Oil	27	17.5	37
Polychlorotrifluoroethylene	38	16.1	37
5-phenyl-4-ether	54	27.8	37

* α-value higher at low shear rates

Table 2 Typical Pressure and Temperature Coefficients of Viscosity for a Range of Fluids

Naphthenic Mineral Oil	0.075
Paraffinic Mineral Oil	0.055
Bicyclohexylether	0.095
Cyclohexyldodecylethe	0.04

Table 3 Typical Traction Coefficients of Fluids (39)

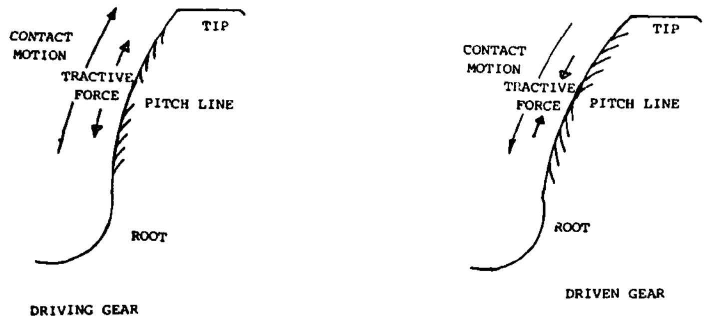

Figure 1 Cracks due to micropitting in involute gear teeth (the size of the cracks is exaggerated). The long solid-headed arrows show the motion of the contact relative to the cracked surface and the short open-headed arrows the direction of the tractive force on this surface.

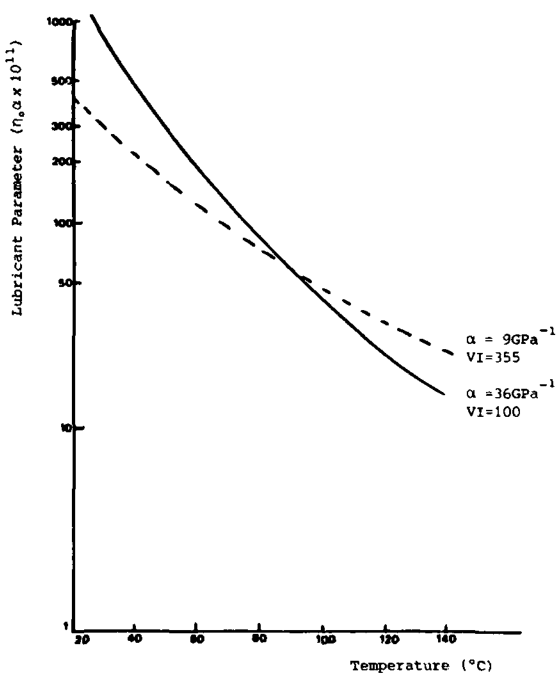

Figure 2 Effect of Temperature on Lubricant Parameter for Two Perfluoropolyethers with Viscosities at 20°C of 250cS (25)

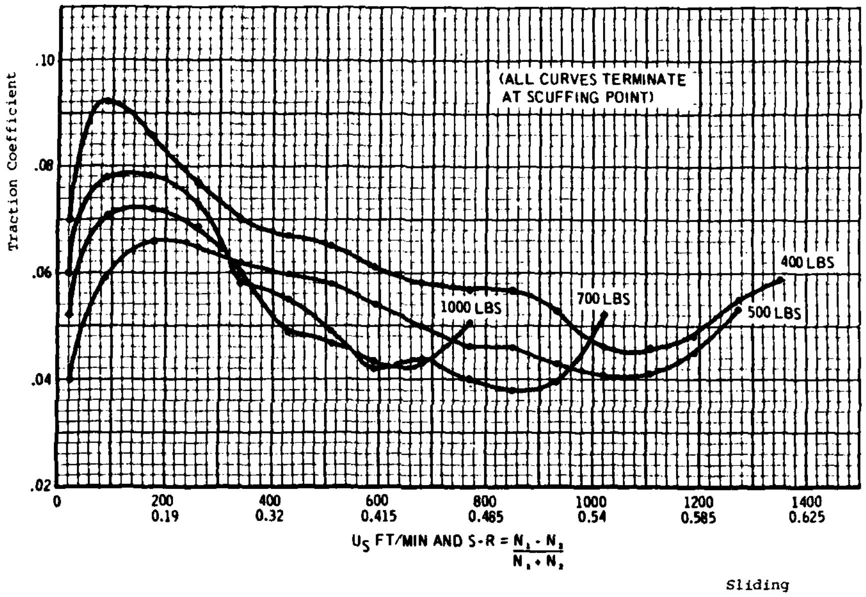

Figure 3 Effect of Slide-Roll Ratio on Traction Coefficient (disc machine, paraffinic mineral oil)

AD-P005 067

AIRCRAFT ENGINE OILS AND THEIR BEHAVIOUR AT HIGH TEMPERATURES

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SUMMARY

Characteristics of aero-engine oils are changed considerably under high-temperature conditions (> 200-250°C). Consequences of this are the formation of aging products and deposits, deficiences in the tribological behaviour and spontaneous ignition, in extreme cases. The problems arising from these are illustrated using several selected examples, and their effects on engine functioning are described.

1. INTRODUCTION

With modern high-performance aero-engines, which are already in service or which will be developed in the future, an important role is played by new design concepts and the use of new materials in the search for the means of increasing performance. An unavoidable consequence of this performance increase is a higher temperature level in the engine; and in addition to the materials, the consumables used for cooling, such as fuel and engine oil, are particularly affected as a result. The possible consequences of high temperatures on the characteristics of engine oils are discussed below, considering especially the condition of the oils under the effects of high engine temperatures, the effects of changes in the properties of oils on their quality requirements, and the tribological behaviour; here the load-carrying capacity. This is illustrated by some selected research findings, based on fresh and used engine oils to different specifications.

2. CAUSES OF CHANGES IN PROPERTIES OF OILS

Changes in the properties of engine oils in principle can have the following causes:

- Chemical attack (air/oxygen, water, catalytic effects)

- Thermal stress
  Both of these result in a thermally induced, catalytic oxidation reaction.

- Mechanical stress (pressure, friction, shear forces)
  Its effect is indirect in that the resultant temperature increase accelerates oxidation reactions.

The temperature to which an engine oil is subjected depends on the engine and flight conditions (type of engine, flying speed and mission etc.). As an indication, at low and high Mach numbers respectively, one can quote figures of 150°C max. and approx 200°C oil temperature in the oil tank (long residence time), and max. 250°C and 300-350°C in the rear bearing chamber.

The influence of increased temperature is extensive:

- Chemical reactions - be they oxidation, decomposition or catalytic reactions - are accelerated (speed of reaction doubles with every 10°C increase in temperature)

- Properties of the engine oils, such as viscosity, vaporisation/vapour pressure, surface tension, foaming etc. are usually impaired

- Finally, the tribological characteristics undergo change

Concerning the temperatures occurring in engine oil systems, the consequences depend on whether the thermal stress is of long or short duration. In this respect, the design of the oil system acquires special significance in that it should be ensured that overheating of the oil is avoided by keeping the residence-times in hot sections of the engine to a minimum[1]. The quantity of the oil, its rate of circulation, the mode of lubrication and the oil consumption are some of the factors that determine the temperature level in the engine oil system. To the effects of these temperatures are added those of

- The temperatures in the bearing chambers near the combustion chamber

- The temperatures at "hot spots", that is to say parts of the engine oil system that are insufficiently thermally insulated or are difficult to cool

- The temperatures at points in the engine oil system in the vicinity of hot engine components where there is little oil or oil/air movement (recirculation areas, oil "sumps" of widely differing dimensions)

- The generally high temperature conditions prevailing at high speeds (above Mach 1), especially at high Mach numbers at high altitudes,

which are far more critical with regard to changes in the properties of an oil.

3. EFFECTS OF CHANGES IN PROPERTIES OF ENGINE OILS

3.1 CHANGES IN CHEMICAL COMPOSITION UNDER EFFECTS OF TEMPERATURE

Even after a short time under normal temperature conditions changes in the composition of an engine oil can already be detected. Taking the chemical changes alone, extensive analytical investigations into laboratory-aged specimens have shown that complex reactions take place simultaneously. Oxidation products of differing molecular weight occur, commencing with volatile and ending with higher molecular products (acids, alcohols, esters, ketones, aldehydes, polymer products, etc.).

As described, the extent of the chemical reaction depends on the temperature and length of time to which the oil is subjected to it, as well as on the volume and temperature of the air. The following stages are passed through with increasing temperature:

- Stage 1
  Aging of the oil is retarded by the effect of additives.

- Stage 2
  Increased formation of volatile products and high molecular, but still partially oil-soluble compounds[2,3].

- Stage 3
  Formation of solid deposits, which is largely governed by length of oil residence-times in engine high temperature areas.

- Stage 4
  Spontaneous ignition of the engine oil.

The last three points and, particularly, their effects on the engine are discussed below.

3.1.1 PROPERTIES OF GREATLY-AGED ENGINE OILS

The first visible sign of changes in the properties of an oil, besides the increase in the depth of colour, is the formation of polymer products, which markedly increase the viscosity. Attempts have been made to analyse these higher molecular compounds (MW up to 50,000), but positive identification lies in the future[4]. The next step is a further polymerisation, recognisable by the formation of sludge or other compounds resoluble in the hot engine oil. Knowledge of their chemical composition is very limited.

3.1.2 SOLID DEPOSITS IN OIL SYSTEM

The formation of solid deposits occurs to a greater or lesser extent in the oil system of every aero-engine. The deposits can give rise to serious problems depending on their amount and location. They can occur in widely differing form from thin layers soft in consistency to thick, hard crusts. Examples by way of scanning electron microscopy can be seen in Fig. 1-3.

Further to visual or microscopic examination, chemical analysis or the determination of physical properties can be used for characterisation of the deposits:

- Analysis of the elements (C, H, N, P, metals[5])

- Infrared analysis (nature of bonds between the elements C, H, N, O etc.)

- Hardness measurements (so far possible only with layered deposits)

- Thickness of layers (total or of individual layers)

In a large number of cases the deposits from aircraft engines can be categorised on the basis of the visual examination and analysis:

- Paint-like, sometimes glossy layers

- Dull layers with a smooth to porous soot-like to grainy compacted structure
- Irregularly formed, dull or glossy particles

More precise classification would be desirable, but this requires further investigations on specimens from engines as well as laboratory tests for simulating deposits occurring in the engine.

The effects of deposits are many and wide-ranging. As long as they remain localised, the following disadvantages must be expected:

- Obstruction of heat exchange, depending on the nature and thickness of the deposit
- Formation of corrosion
  Corrosive attack occurs beneath the deposits and is dependent on the material used. It is caused by greatly aged oil with high acid numbers (several mg KOH/g oil) which is found in the deposits, especially those with porous structure.
- Preliminary damage to bearings and gears is initiated when overrolling of the deposit occurs.

The situation is more serious, because damage can develop rapidly, when the oil deposits burst. They can then clog filters or, if ground down sufficiently, block oil jets.

Information about conditions at which deposits are formed can be gained, firstly, from measurements made in the course of rig tests of engines (subject to appropriate instrumentation) and, secondly, from laboratory tests, in which the flexibility and simplicity of the instruments and apparatus permits them to be used in a number of ways. Laboratory test apparatus differs, on the one hand, according to the manner in which the oil is used - as a film, vapour or droplets - and, on the other hand, according to design (from laboratory equipment to test rigs)[6-8].

Suitable techniques must be selected for testing the propensity of oils to form deposits. This should be clarified by the following example using various ester-based engine oils and additives. Testing was carried out using a thin-film tester and a panel coker at the DFVLR (German Aerospace Research Establishment, Stuttgart), both of which are characterised by the high reproducibility of their measurements[9].

The oils investigated were three oils approved according to the British specification DERD 2497 ($\eta = 5mm^2/s$) and an oil with $\eta = 4mm^2/s$. In testing with the thin-film tester the low-viscosity oil proved outstanding (Fig. 4). On the other hand, in the panel-coker test the high-viscosity oils proved best.

Judging solely by these results one might easily dismiss the method of testing as being irrelevant. If one looks at the test apparatus in detail one comes to a plausible explanation. In the test with the thin-film tester, a specific amount of oil is applied once only to a metal plate, which is heated to the selected test temperature according to a preset schedule. On conclusion of the test the oil residue is weighed. In the panel-coker test, whiskers affixed to a rotating shaft fling oil from an oil sump onto a hot panel, where used oil is constantly replaced by fresh oil (Fig. 5). Here also, the oil residue adhering to the panel is weighed on completion of the test.

The difference between the two methods lies in that the used or evaporated oil is replaced in the panel-coker test, which means that the quantity of oil on the hot panel remains constant for the duration of the test.

Determination of the evaporation losses shows clearly that the oil with $\eta = 4mm^2/s$ has a very low tendency to form deposits at the test temperatures in question, but exhibits high evaporation losses (Fig. 6).

This example clearly shows the influence of the test apparatus used and highlights the need to identify what type of deposit lies where in the engine to define which simulation tests should be carried out.

3.1.3 SPONTANEOUS IGNITION OF ENGINE OILS

As the temperatures in the engine oil system climb even higher (both locally and generally) the greater becomes the danger of spontaneous ignition of the oil. Here we are not concerned with situations in which contact between two bodies can generate sparks, leading to ignition of the oil/air mixture at relativel low temperatures (150°C).

In principle, the tendency to spontaneous ignition of engine oils is favoured by the following major factors:

- Oil vapour/air-mixture ratio close to stoichiometric conditions
- High temperatures (> 300°C)
- Protracted residence-times
- Mixture (design influence)

As is known, these factors can be influenced by design measures and by the dimensioning of the oil/air flow such that spontaneous ignition of the engine oil is prevented. However, spontaneous ignition can occur when there is a combination of factors such as extreme flight conditions, seal-wear resulting in changes in the oil/air mixture ratio, high performance etc. Fire of varying seriousness and duration can then break out, from oil flash up to heavy fires. The extent of the ensuing damage and consequences for the operation of the aircraft engine depend on the nature of the fire and its location in the oil system.

As demonstrated in rig tests, there is no detectable damage when the ignition of the oil is of short duration (lasting just for a few seconds). Usually the oil/air mixture is so rich that the ignition threshold is not exceeded. However, if spontaneous ignition does occur at individual places as a result of leaner mixture, the combustion will be incomplete and considerable quantities of combustion residues will be formed. These will pass through the oil system and will become trapped in the filter or deposited in cavities in the oil system; in which case the number, i.e. the extent, of the oil fires determines whether detrimental effects (similar to the effects of deposits) are to be expected.

Very serious damage occurs when an oil fire becomes established at a component, where the high temperature of the fire (1000°C plus) can lead to softening of the material and changes in its properties. To gain a better understanding of the sequence of events with spontaneous ignition and to ascertain the main influencing parameters, theoretical studies on the basis of the heat absorbed and dissipated by the system in question were carried out. Confirmation of the findings was then sought in laboratory tests using different types of equipment. In the first place, this was a matter of simple, standardised (DIN/ASTM) apparatus suitable merely for comparing the spontaneous ignition temperatures of various oils. In the second place, apparatus was developed for laboratory-simulation of the processes leading to the ignition of flammable liquids in engines and other industrial machines, for gaining knowledge about the combustion characteristics. These studies then provide information regarding safety engineering[10,11].

The various tests, carried out under static or dynamic conditions, concerned the following parameters, which influence the spontaneous ignition behaviour:

- Oil/air mixture ratio
- Temperature (oil, air, wall)
- Residence-time
- Volume to surface ratio
- Oil state (vapour, droplets, film)
- Ignition delay
- Pressure
- Chemical composition
- Surface material
- Roughness

Few laboratory investigations into the spontaneous ignition characteristics (not external ignition) of aero-engine oils in relation to the engine oil system have been carried out. Depending on whether the tests are conducted under static or dynamic conditions, the results vary widely. The bottom ignition temperature e.g. for ester based oils (0-160) lies at 280°C (BAM bomb test method using an injected oil/air mixture of stoichiometric composition). Other values for oils to different specifications and based on other test methods[12-14] are listed in the table in Fig. 7.

The results of our own tests using the BAM bomb method concerning the relationship between the spontaneous ignition temperature and pressure of three 0-160 oils are shown in Fig. 8. The slight difference in the ignition temperatures and the slight dependence on pressure of the three oils is striking. Bigger differences using mineral oil or phosphate ester lubricants are known from the literature[15].

One aspect that has not been investigated to any great extent so far is the effect of aging of the oil; all research has been carried out using fresh oil. Initial tests using a simple simulator show that oxidative aging of ester oils gives rise to evaporable products which have a lower spontaneous ignition temperature than the engine oil. These products, which occur when air is passed through oil at temperatures > 250°C and which condense out of the airstream, in addition to low-boiling oxidised products, contain acids in the carbon atom range of 5-10. The spontaneous ignition temperature decreases with increasing chain length (number of carbon atoms). The extent to which spontaneous ignition is initiated by these products depends on the extent to which they are formed in the engine. The limiting factor is a concentration which leads to overshooting of the leaner ignition threshold.

3.1.4 LOAD-CARRYING CAPACITY OF OILS AT ELEVATED TEMPERATURE

Testing under mechanical load also belongs to the methods used for evaluating the quality of an oil. To improve the load-carrying capacity of an oil under the influence of external forces, additives are used, and these can undergo change or be consumed during service. Changes at high temperatures in the engine may well be far-reaching. Of course, there are now sensitive analytical procedures available (chromatography, mass spectrometry) for determining the extent to which the various additives for improved load-carrying capacity become exhausted in used oils[16]. However, so far, there is no correlation between the individual analysis results and the actual residual load-carrying capacity of a used oil.

Investigation of the load-carrying capacity of an oil in a laboratory or rig test always suffers from the disadvantage that the tribological conditions encountered in practice cannot be simulated or can only be approximated. When testing the load-carrying capacity of oils one must choose between laboratory apparatus (inexpensive, large number of samples, small oil quantity, little resemblance to practice) and more elaborate test-rig apparatus (expensive, small number of samples, large oil quantity, moderate to good simulation of practice conditions).

In the research into the high-temperature characteristics of engine oils, test runs were carried out on the gear test rig at FZG (Laboratory of Gear Research, TU MÜNCHEN) using oil samples from a modern combat aircraft. The starting quality of the lubricating oils to DERD 2497 (0-160) used was investigated on an FZG rig in a test under normal conditions to DIN 51354, as well as in a test programme under excessive temperature and rotation conditions. The results are represented in Fig. 9; for sake of comparison, the values for oils to other specifications, tested by FZG, are also entered. The good load-carrying capacity of the DERD 2497-oils in comparison with the other oils is noticeable, as also is the uniform load-carrying capacity of the B, in both test programmes. The results are attributable to the varying additives and base-oil compositions.

Other tests were carried out using oils from test bed engines as follows:

- Oil A to DERD 2497; high-temperature test (simulation of high Mach numbers and great altitudes)
- Oil B to DERD 2497; running-in phase (total of 4 hours engine running time)
- Oil B to DERD 2497; high-temperature test (simulation of high Mach numbers)

For a total volume of 15 l, the oil consumption was approx. 0.5 l/h, under high temperature conditions 1 l/h or more. Analysis of the individual oil samples reveals that distinct changes in viscosity and acid number as well as a heavy drop in additive content can only be determined with oil A (Fig. 10).

The results of the tests on the FZG rig (test under excessive conditions) are illustrated in Fig. 11.

Two results are striking:

- The two high-temperature tests (oils A and B) result in a greater diminution of the load-carrying capacity than occurs in the running-in test where high mechanical stress is to be expected.
- Although oil B exhibits particularly good load-carrying characteristics, in the high-temperature test used oil B exhibits the same reduced load-carrying capacity as the "poor" oil A.

It cannot be concluded from these results that high temperatures greatly diminish load-carrying capacity. The findings could also be explained as being related to engine running time (100-150h in both cases A and B). Without going into the differences in the parameters, such as pitch line velocity, gearing and oil temperature, of the FZG gear-oil test and the engines in detail (see Ref. 17-19); the findings of the inspection concerning the engines from which oil samples A and B were drawn are reported on.

In neither case were there demonstrable signs of serious damage, either to the gears or bearings. However, there is a certain degree of uncertainty about this, since the engines were run for a short time only (100-150 h) and incipient damage in rotating parts is hard to detect with simple test equipment.

From the findings it may be concluded that, with the engine concerned, a used oil at load stage 5 still does not lead to significant damage, and that high temperatures do not have a striking influence on the load-carrying capacity. Possibly the lower load carrying capacity due to diminution of the additive content under the influence of high temperatures can be equalised by other temperature effects. For instance oil consumption is increased at high temperature engine runs resulting in large amounts of fresh topped-up oil. Or the degradation products of the engine oil such as organic acids are known to improve the load-carrying capacity.

In short, it can be said - as the example shows - that reduced load-carrying capacity does not necessarily result from high thermal stress. To what extent this finding may be generally applied depends mainly on the engine type considered.

4. SUMMARY

Under the influence of the high temperature encountered in modern high-performance jet engines, and probably even more so in future engines in which the temperatures are expected to be even higher, degradation of the properties of engine oils results in deficiencies. These deficiencies have their origin in reaction products of the oil, in spontaneous ignition of the oil and to a certain extent in reduced mechanical loading capacity.

These deficiencies and their effects are highlighted on the basis of the results of investigations into fresh and used oils.

At present, one of the most serious problems, if spontaneous ignition is regarded as an isolated phenomenon, is that of deposits, because deposits are formed by irreversible reactions, i.e. even changing the oil will not have any appreciable effect. Although highly thermal-stressed oils showed a loss in load-carrying capacity, the influence of temperature is yet to be clarified.

All in all, it would seem that the thermal stress capacity of engine oils has been largely exhausted, meaning that further development of the oils first of all signifies an improvement in the thermo-oxidative stability and low coking propensity.

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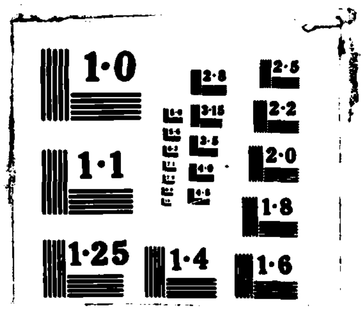
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Fig. 1 SEM-picture of a deposit from aero-engine (paint-like layers) magnification x270

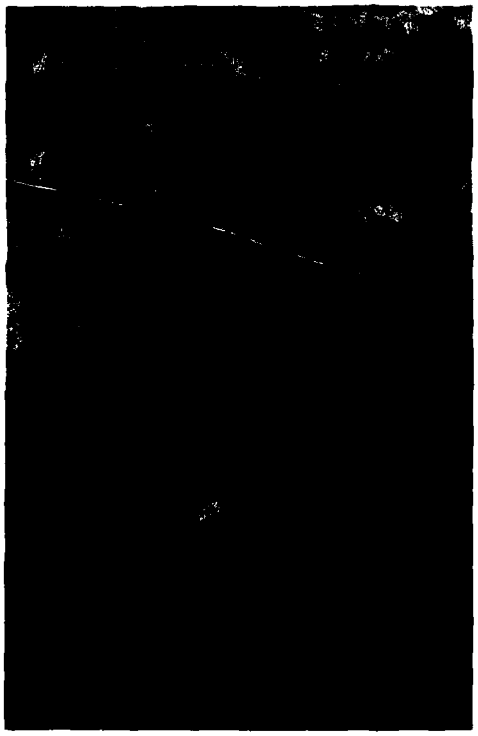

Fig. 2 SEM-Picture of a deposit from aero-engine (dull, soot-like layer) magnification x500

Fig. 3 SEM-picture of a deposit from aero-engine (irregularly formed particles) magnification x400

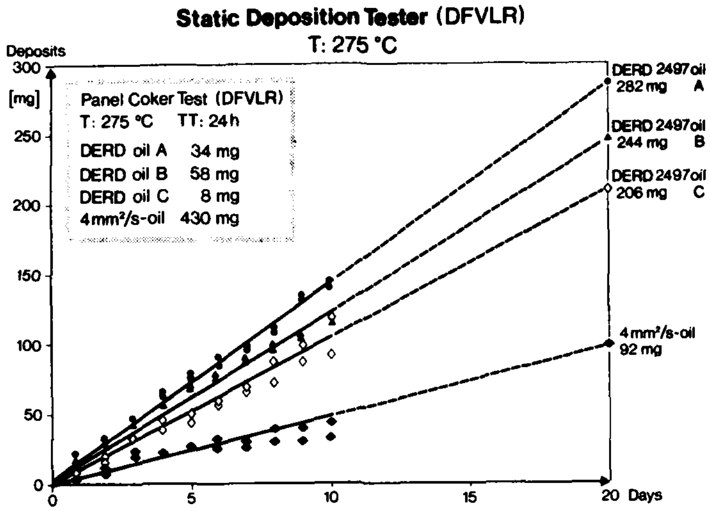

Fig. 4 Results of deposition testing with different engine oils (η = 4 or 5 mm^2/s)

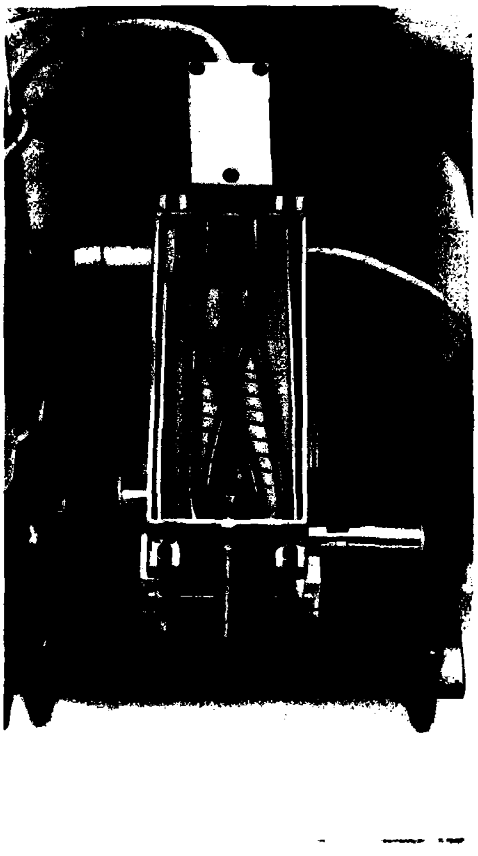

Fig. 5 View on the Panel Coker tester (DFVLR)

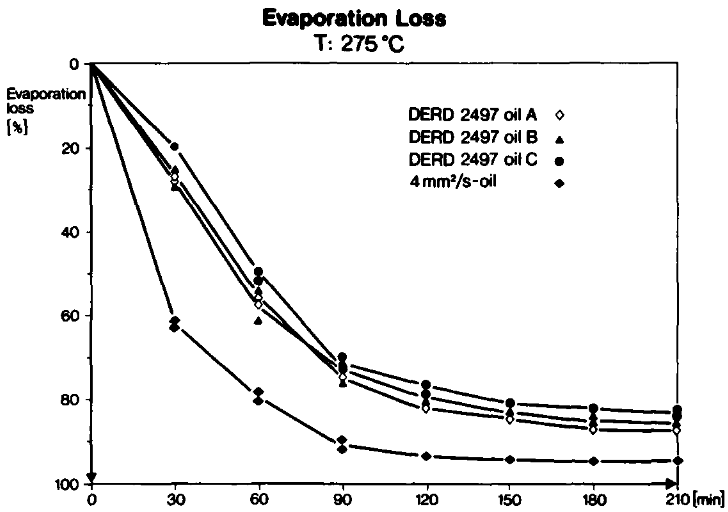

Fig. 6 Results of evaporation loss determination of different engine oils (η= 4 or 5 mm²/s)

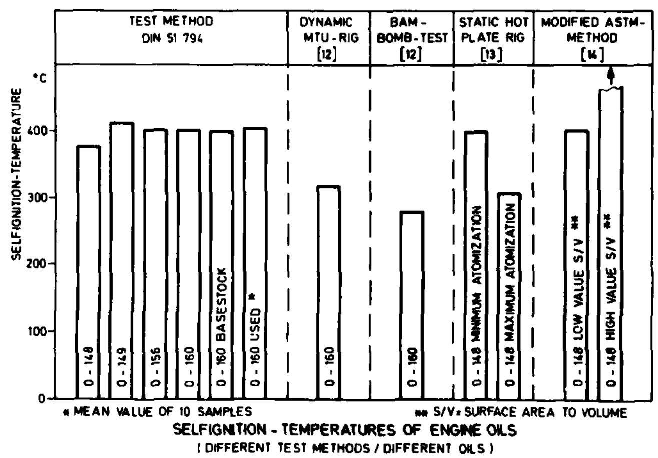

Fig. 7 Selfignition-temperatures of various engine oils

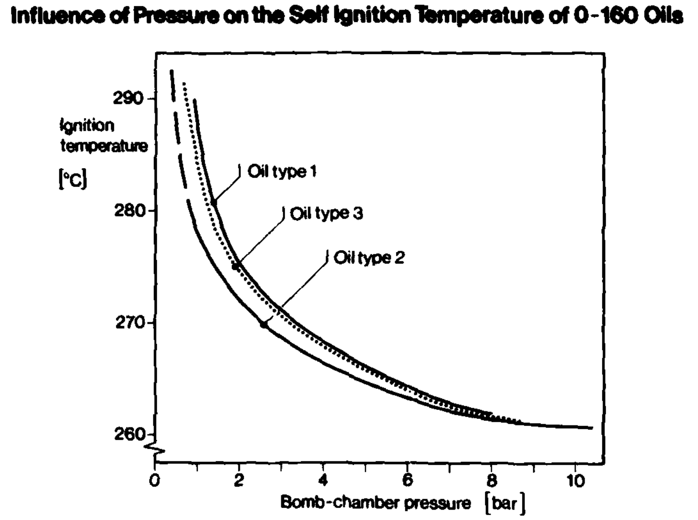

Fig. 8 Variation of selfignition-temperature with pressure raise (DERD 2497-oils)

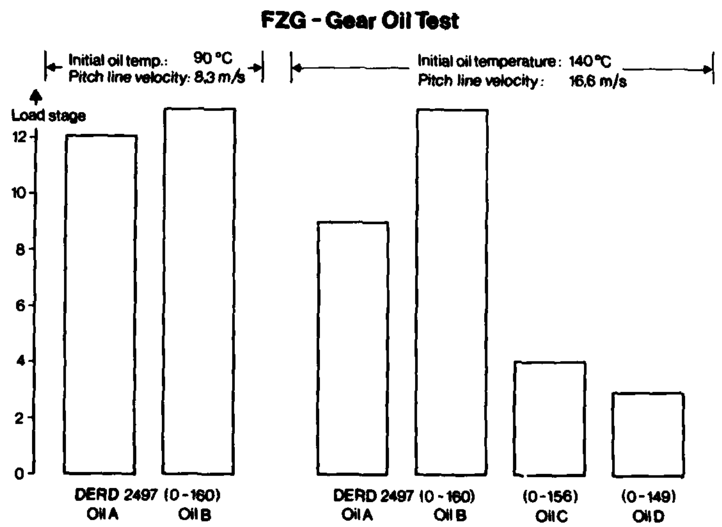

Fig. 9 FZG-gear oil test: Results of engine oils to different specifications

Oil Analysis Results

	Simulation of engine conditions	Viscosity at 40°C mm^2/s	Total acid number mg KOH/g oil	Additive content %		
				Antioxidans		Load carrying capacity
				Typ A	Typ B	
Oil A	Great altitudes, high mach no.	28.60	2.06	42	0	38
Oil A	— (unused oil)	26.90	0.64	100	100	100
Oil B	4h run-in test	26.12	0.51	97	96	90
Oil B	High mach no.	27.40	0.57	97	48	76
Oil B	— (unused oil)	26.10	0.40	100	100	100

Fig. 10 Comparison of oil analysis results of used engine oils and unused oils (DERD 2497)

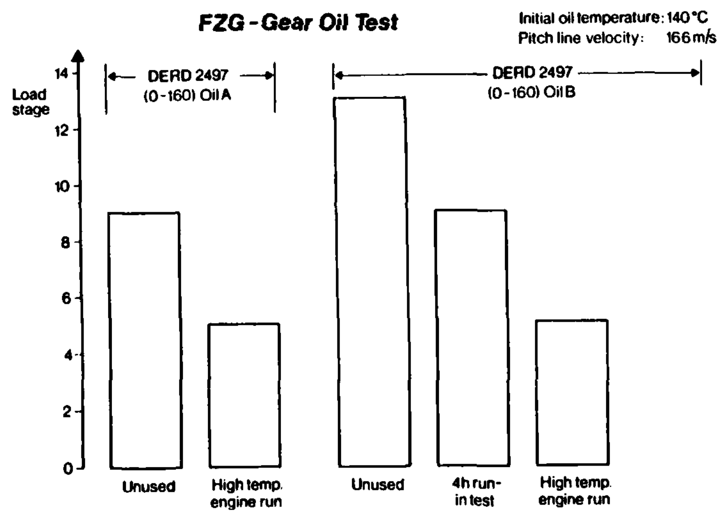

Fig. 11 FZG-gear oil test: Results of used engine oils (DERD 2497)

Performance Modelling-A Tool For Lubricant Development

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SUMMARY

A basic requirement for the development of lubricants and aerespace fluids is to understand the physical properties of these fluids required to optimize performance of the lubricated system. Computer models have been used to predict the effects of lubricant properties on rolling element bearing performance. Studies were performed to validate models, to predict performance under various operating conditions, and to predict the effects of proposed design or lubricant changes on system performance. The results of some of these studies are described and compared with experimental measurements. Models which predict traction forces in lubricated contacts from thermal and rheological properties of the lubricant and bearing materials, and the geometry and operating conditions of the contact are compared with experimental results from a two disk traction device with a micro-computer based control/data acquisition system. The capabilities of models with the ability to predict the performance of hydraulic systems, the components making up these systems and lubricated contacts within the components are described and the methodology of studies to determine required fluid properties discussed.

INTRODUCTION

Computerized mathematical models, used to predict performance, have advanced developments in areas such as airframe structures and aerodynamics for many years. Such models can reduce the time and expense needed to conduct tests to validate designs, allow an understanding of the behavior of experimentally unobservable variables, and lessen the necessity of building prototypes to assess behavior of proposed systems. The development of mathematical models of the performance of fluids and lubricants and the role such models are expected to play in the development of these materials for Air Force applications is described in this paper.

Interest in mathematical models of lubricant performance began with attempts to model the performance of rolling element bearings on the mainshafts of turbine engines and in spacecraft attitude control systems. For turbine engine bearings, the objective of this modelling was to improve the design of bearings to reduce the level of forces acting between the bearing elements and to reduce excessive wear in the bearing. In the case of bearings in spacecraft attitude control and pointing mechanisms, the objective was to enhance understanding of the lubrication process so that methods could be developed to reduce torque anomalies and predict the expected operational life of these assemblies. Dynamic models of rolling element bearing performance have been applied to improve designs of existing liquid lubricated bearings and have been used to improve the understanding of the performance of experimental solid lubricated bearings for high temperature applications.

From numerous studies of bearing performance using mathematical models, it has been learned that the traction in the lubricated contacts between bearing elements and its variation with the sliding velocity in the contact has a strong effect on bearing performance. This force is influenced, among other things, by the nature of the lubricating material. An experimental device in which the perimeters of two disks are loaded against each other has been employed by AFWAL to make experimental measurements of the traction force for various lubricating materials. These results are used as inputs to mathematical models of lubricated systems employing these materials, to deduce information about the physical properties of the materials, and to validate models relating the traction force to the physical properties of the fluid.

The traction force in lubricated contacts primarily arises from the resistance to shear of the lubricating film separating the lubricated surfaces. Given a suitable model, it should be possible to relate traction measurements to fundamental physical and chemical properties of the lubricant. A model based on the assumption that lubricants behave as Newtonian fluids has been used to analyze traction data gathered at the Materials Laboratory and determine a viscosity, pressure-viscosity coefficient, and temperature-viscosity coefficient at ambient conditions and at conditions representative of the region within the lubricated contact. The coefficients so determined are found to disagree with measured viscosity data at pressure and temperature for lubricants. Models based on more accurate representation of the fluid behavior are being developed.

An activity of the Materials Laboratory is the development of new hydraulic fluids which are more fire resistant and suitable for higher temperature applications than those materials now being used. These materials are of higher density than materials presently being used and, to reduce the weight of hydraulic systems using these materials, systems using higher operating pressures are being developed. Many lubrication related failures have occurred in tests of these systems. Lubrication modelling techniques available to examine these failures are discussed in the paper.

The application of lubrication models is a complex process, involving many steps beyond development of the model. Modelling techniques have been applied to the performance of lubricants in bearings, to lubricant traction modelling, and to the performance of hydraulic fluids. They have led to a more effective understanding of the lubrication process and to the development of better performing lubricants.

BEARING PERFORMANCE MODELLING

Rolling element bearings are machine elements which provide a means of transmitting loads between a rotating shaft and the stationary structure of the machine. One of several types of rolling element bearings, a ball bearing, is shown in Figure 1. It is made up of three types of elements typical of all rolling element bearings, which come in many specialized configurations. The rolling elements (balls) transmit the load and by rotating provide an interface with the rotating shaft and the stationary structure in which sliding velocities are low, thus minimizing wear, power loss and friction. The inner and outer race are mounted on the shaft and in the structure respectively and are grooved to provide a path of motion for the rolling elements. The retainer maintains proper spacing between bearing elements. The first computerized models of bearing performance were quasi-static models developed by Jones[1,2] which, in addition to static forces in bearing systems, accounted for the centrifugal and gyroscopic forces arising from the accelerations of the bearing rolling elements. Such models simulated the steady state interactions between the rolling elements and the bearing races reasonably well. The first model which accounted for the interactions of the other bearing elements with the retainer and for dynamic changes in bearing element motions was developed by Walters[3]. Gupta developed the first truly dynamic bearing model[4,5] and later simplified this model to reduce computation time[6]. Further development of bearing models[7,8,9,10] is driven by the need to reduce computation time, to simulate particular configurations, and to study specific phenomena affecting bearing performance.

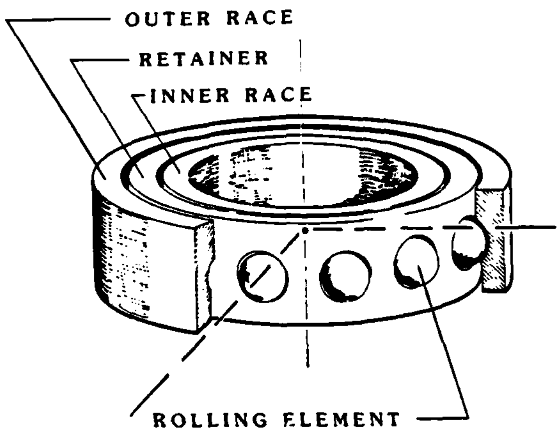


Figure 1. Elements of a Rolling Element Bearing

The Air Force Wright Aeronautical Laboratories (AFWAL) have furthered the development of bearing modelling as a technique of lubrication analysis by sponsoring development of models, by validating the models through correlation of model predictions with experimental results, and by applying the models to improve the design of operational and experimental bearings and to provide an improved understanding of the function of lubricants in bearing performance. Each of these is an essential step in making computerized mathematical models practical for use in the analysis of lubrication systems.

An early Air Force application was to use the DREB[5] program to study the performance of a bearing proposed for use in a Reaction Wheel Assembly (RWA), a device used for stabilization and control of satellites. A similar bearing had experienced problems under certain operational conditions. The approach used was to simulate the performance of the bearing both under nominal operating conditions and under conditions where problems were experienced. Further, parametric studies were conducted to study the performance of the bearing with changes in retainer design. Figure 2 shows the motion of the mass center of the bearing retainer within the clearance limits of the bearing for three different sets of operating conditions. It may be seen that under nominal operating conditions the movement remains confined to the region near the center of the clearance circle. A lower film thickness causes the retainer to move toward the clearance limit, and with the added complication of low temperature, the interactions between the retainer and the races become much more frequent and involve forces of sufficient magnitude to significantly affect bearing torque. Figure 3 shows the radial position of the retainer mass center and the resulting bearing power loss for several values of retainer ball pocket clearance. It may be seen that with increased ball pocket clearance, the retainer quickly assumes motion at a constant radial position and power loss declines significantly. Based on studies of this kind, it was recommended that the retainer ball pockets be enlarged in the bearing selected for this design. Initial tests showed improved performance with this modification. This study gave confidence in the capability of models to predict bearing performance and led to an improved retainer design which was selected for operational use.

Within the Air Force, significant studies have been directed toward the study of turbine engine mainshaft bearings. These studies have had the objectives of validating the models for these applications, of investigating the effects of shaft imbalance on impact forces on the bearing retainer, and improving the design of bearing retainers. To obtain data to validate the capability of the model to simulate retainer motion in turbine engine bearings, an experimental program[11] was conducted in which a bearing test rig was modified so that proximity and photonic probes were mounted relative to the retainer surface as shown in Figure 4 in order to sense the retainer linear and angular displacements and its angular velocity about its axis of rotation. The test device was capable of spinning the shaft at 30,000 RPM and had provisions for heating and circulation of the lubricant to simulate engine bearing cavity conditions. Figure 5 illustrates the correlation between experimental and analytical results by showing the retainer angular velocity at several shaft speeds as predicted by the model and as

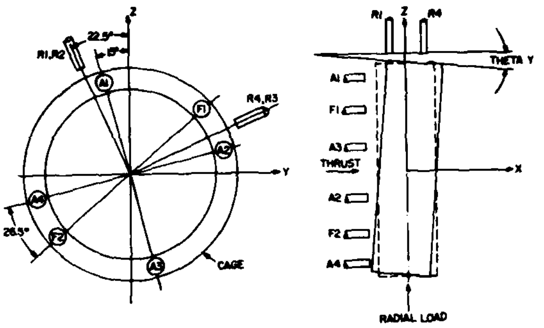


Figure 4. Probes Directed at Retainer Surface

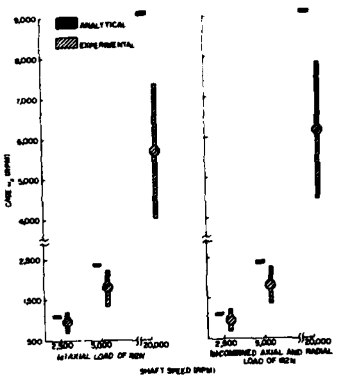
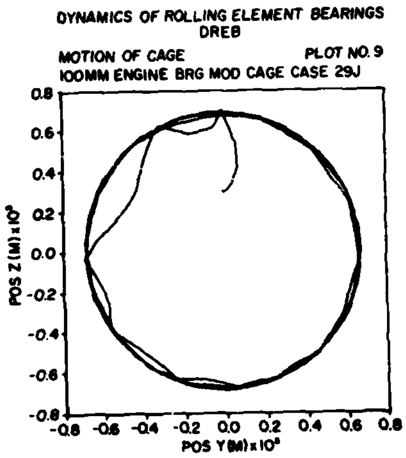


Figure 5. Retainer Angular Velocity Thrust and Thrust/Radial Load



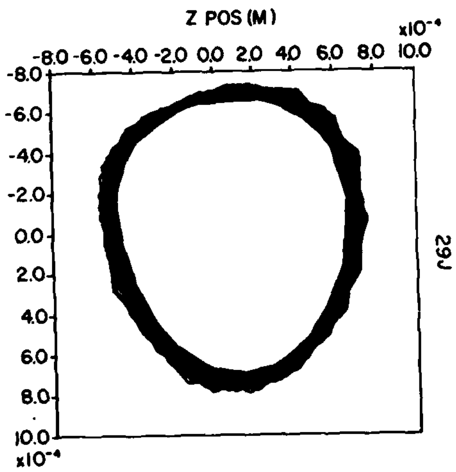


Figure 6. RAPIDREB Simulated Orbit
Retainer Mass Center Orbits, Thrust Load=Axial Load = 1112 N, Shaft Speed = 20 KRPM

Figure 7. Experimental Orbit

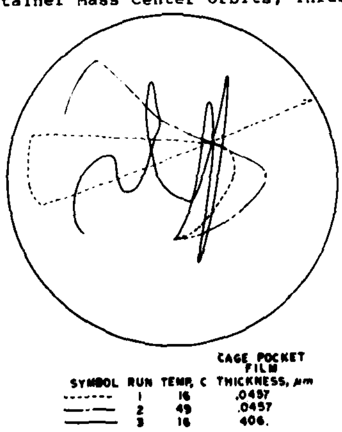


Figure 2. Retainer Mass Center Orbits

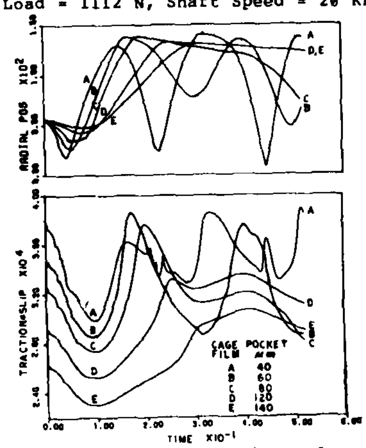


Figure 3. Retainer Motion and Power

experimentally measured. Both the model and experimental results showed an orbital motion of the retainer about the clearance limits at high speeds, as shown in Figures 6 and 7. It was shown, using the models capability, that this motion is assumed at a critical speed dependent upon the friction characteristics in the contacts between the bearing elements.

Several studies of the performance of solid lubricated rolling element bearings have shown the dependence of proper design on the frictional characteristics of the lubricant employed. A parametric study[12] of an experimental bearing was performed in an attempt to provide guidance in the design of a self-lubricating retainer. Based on assumed frictional characteristics, performance was simulated over a matrix of design parameters as shown in Table 1. Figure 8 shows the power loss in the bearing for two values of retainer ball pocket clearance. The power loss is seen to be composed of a nominal value and an additional increment which occurs during collisions between the bearing elements. Both the magnitude and frequency of the incremental power loss is dependent on both the clearances between bearing elements and the frictional characteristics in the bearing contacts. Figure 9 shows the impact force between the retainer and race for two values of clearance between the retainer and race. It may be seen that a significant coning motion, indicated by higher impact forces on one race, exists in the bearing. This would lead to preferential wear of the retainer on one side, which may in turn cause an imbalance, aggravating any tendency toward instability in the retainer motion.

Table 1. Run Numbers For Parametric Bearing Geometry Effects Study

Retainer/Race Clearance mm	Guidance on Race	Retainer Ball Pocket Clearance, mm			
		0.60	0.4572	0.30	0.20
		Parametric Study Run Number			
0.60	Outer	1.1	1.2	1.3	1.4
0.40	Outer	2.1	2.2	2.3	2.4
0.3302	Outer	3.1	3.2	3.3	3.4
0.20	Outer	4.1	4.2	4.3	4.4
0.60	Inner	5.1	5.2	5.3	5.4
0.40	Inner	6.1	6.2	6.3	6.4
0.3302	Inner	7.1	7.2	7.3	7.4
0.20	Inner	8.1	8.2	8.3	8.4

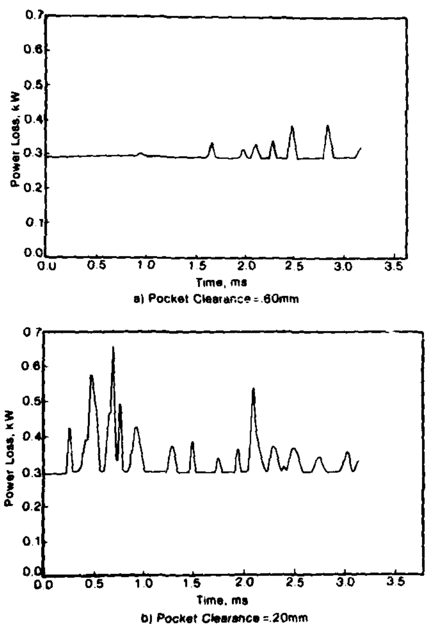


Figure 8. Total Bearing Power Loss
Race Clearance 0.3302 mm

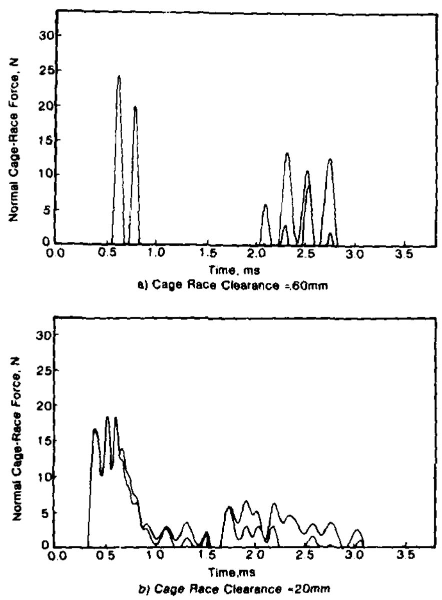


Figure 9. Retainer-Ball Impact Force
Pocket Clearance 0.20 mm

In another study of solid lubricated ball bearings[13], experimental measurements of the friction coeffient with various combinations of solid lubricant, steel, and ceramic were used as inputs to a model of the bearing performance. Table 2 is a summary of the test parameters used in this program which represented sliding speeds, temperatures, loads, and material pairs present at various contacts in this hybrid bearing with ceramic

balls, steel races, and a retainer made of a self lubricating composite material. It was found that under some conditions collisions involving excessive forces would occur progressively in adjacent ball pockets of the retainer leading to a growth in power loss as depicted in Figure 10. A typical failure of the experimental bearing is depicted by the retainer shown in Figure 11, in which a web between adjacent ball pockets has been sheared by excessive ball-retainer forces. This is the type of failure predicted by the model study.

Table 2. Summary of Test Conditions

TEST PARAMETER		MATERIAL TEST - PAIRS		
		M-50 Steel vs. HAC-1 (1)	Silicon Nitride NC-132 vs. HAC-1 (1)	Silicon Nitride NC-132 vs M-50 Steel (2)
SPEED	rpm	0-30,000	0-30,000	0-50
	cm/sec	0-7,980	0-7,980	0-13.3
AMBIENT TEMPERATURE	°C	204	204, 316	204
	°F	R.T.,400 (3)	R.T.,400, 600	R.T., 600
CONTACT STRESS	MPa	0.69 to 20.7	0.69 to 20.7	690 to 1725
	Kpsi	0.1 to 3	0.1 to 3	100 to 250 (4)

NOTES:
(1) HAC-1/T50 Fl and HAC-1A/T300 Fl.
(2) Lubricated by stick lubricant transfer.
(3) Temperatures limited because of bond to metal strength of HAC materials.
(4) Initial contact stresses.

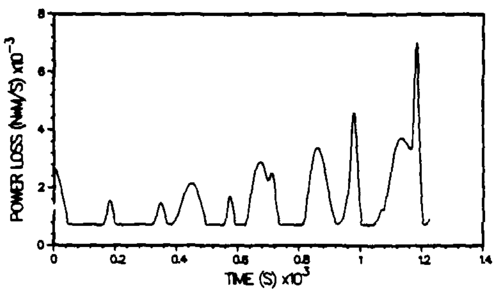


Figure 10. Total Bearing Power Loss

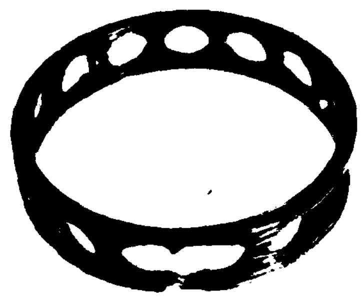


Figure 11. Failed Bearing Retainer

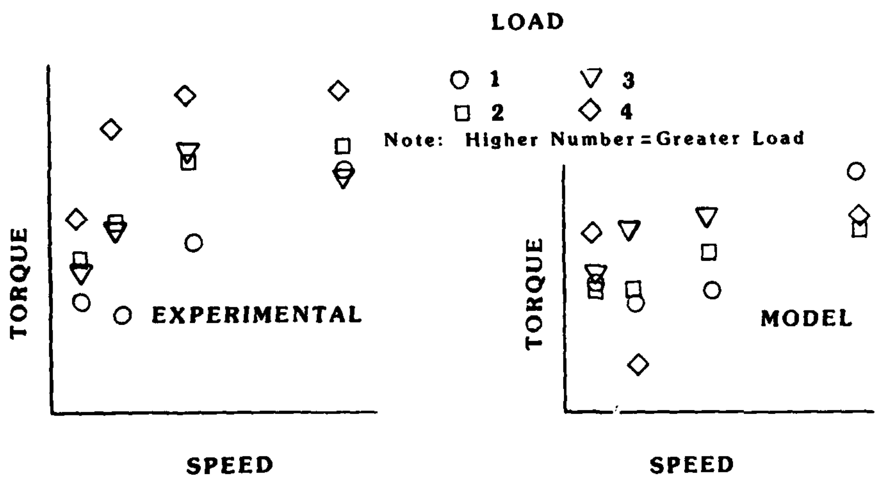


Figure 12. Computed and Experimental Average Bearing Torques vs. Rotational Speed

Experimental studies and simulations using computer models of bearings for use in the despin mechanical assembly (DMA) of satellites have also been conducted. DMAs are devices which form an interface between the body of a spacecraft, which is rotated to provide stability and control of the spacecraft's attitude, and a platform which is despun so that antennas, sensors, and other directional devices can be pointed at fixed targets. The assembly consists of a pair of bearings mounted between a shaft and housing affixed to the platform and spacecraft, a drive motor to control spin rate, and a slipring assembly for the transmission of electrical power and electronic signals between the platform and spacecraft. Experimental tests of isolated bearings and sliprings and of flight quality DMAs have been performed to provide data for correlation with model studies. In Figure 12, experimentally measured values of bearing torque are compared with those predicted by a model of a DMA bearing. The magnitudes vary somewhat between the predicted and computed values because the traction values used for the lubricant were extrapolated from a limited amount of available data. To further assess the success of the model in predicting changes in bearing behavior with speed and load, straight lines were fit through the traction values for each load. The slopes of these lines are shown in Figure 13(a), which shows the derivitive of torque with respect to speed plotted vs. load. An analogous procedure performed on a crossplot of Figure 12 having torques vs. load for various speeds resulted in Figure 13(b), showing the derivitive with respect to load for various speeds. While the absolute values are again different in value, the model is shown to predict bearing behavior changes quite well.

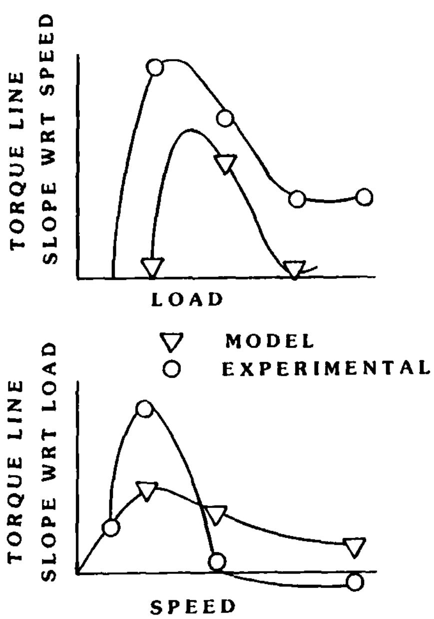


Figure 13. Behavior of Torque Changes

TRACTION OF LUBRICANTS

The bearing modelling studies performed have greatly advanced the understanding of the influence of lubricants on bearing performance. The tractive force transmitted across contacts between lubricated machine elements is an important influence on the temperature, wear, and power dissipation in those contacts. Modelling these forces, even for lubricants with well known physical properties, is a demanding task because of the many variables which must be considered. When lubricants are being developed, the physical properties are usually not well known and the modelling process is further complicated. For these reasons the Materials Laboratory has constructed a unique twin disk traction measurement device[14] to measure these properties of lubricants. This device has several unique features making it suitable for research into the properties of experimental materials. Approximately 250 ml of fluid is needed for testing, thus lubricants may be tested at an early stage of development. The disks are easily changed and inexpensively constructed so that research into regimes such as boundary lubrication or solid lubrication, where disks may be quickly worn, is practical with this machine. A microcomputer based system has been developed[15] to control operation of the machine during an experiment and collect all the data, minimizing manpower requirements for testing and removing human error as an experimental variable.

Materials which have been tested on this device include lubricants for use in rolling element bearings in space applications, in turbine engines, and in electrical motors. Three hydraulic fluids have also been tested. The test conditions shown in Table 3 are indicative of the capabilities of the device. After completion of testing, data is transmitted to a main frame computer for further data reduction, analysis using techniques to be discussed later, and archival storage. This device makes it practical to accumulate a massive data base of traction characteristics of a wide variety of experimental and operational lubricants and is expected to contribute significantly to the advancement of lubricant development.

Table 3. Lubricant/Fluid Traction Tests

LUBRICANT FLUID	TEMPERATURES °C	ROLLING SPEEDS m/s	HERTZIAN CONTACT STRESS GPa
MOBIL M-28 BASE OIL	25, 40, 55, 93	6.88, 8.53, 10.2	0.52, 0.69, 0.86, 1.03
SHELL TURBO 78	25, 40, 55, 93	6.88, 8.53, 10.2	0.52, 0.69, 0.86, 1.03
VAC KOTE 36233	25, 38	0.18, 0.36, 0.71	0.28, 0.30, 0.35, 0.39, 0.43, 0.46, 0.48, 0.49, 0.52, 0.54
MIL-H-83282	93, 116, 135	5.03, 9.73, 14.1	0.52, 0.69, 0.86, 1.03
MIL-H-5606	93, 116, 135	5.03, 9.73, 14.1	0.52, 0.69, 0.86, 1.03
CTFE	93, 116, 135	5.03, 9.73, 14.1	0.52, 0.69, 0.86, 1.03

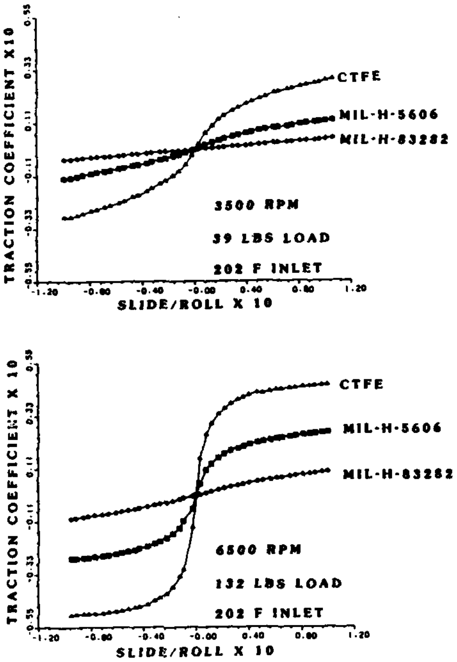


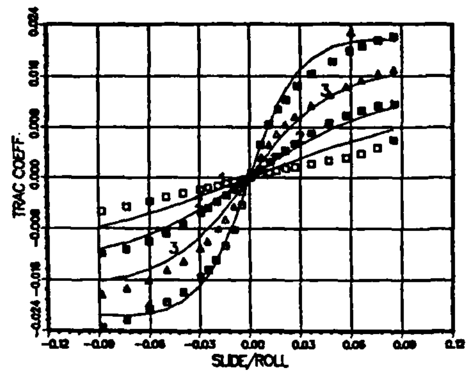
Figure 14. Hydraulic Fluid Tractions

The traction characteristics measured for three hydraulic fluids at two sets of operating conditions are displayed in Figure 14. It may be seen that the effects of changes in load and speed are similar for all the fluids. While the traction coefficient of MIL-H-83282 tends to remain linear with shear rate (slip speed) that of the other fluids shows a pronounced nonlinearity. The experimental fluid, CTFE, shows traction behavior more like that of MIL-H-5606 than the other fluid. An assessment of the significance of these differences awaits more complete analysis.

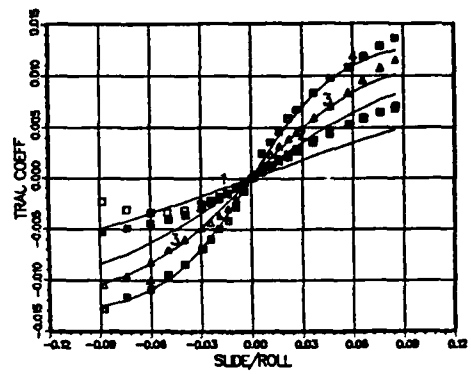
There are several reasons for interest in modelling the traction forces arising from the resistance to shear of the fluid film in a lubricated contact. Such forces should be relatable to fundamental physical properties describing the behavior of the lubricant. Given a suitable model, it should be possible to determine the values of these properties from analysis of data of the nature discussed previously. Traction values are useful and essential inputs to lubrication models of the sort described earlier in this paper and a suitable traction model is a useful tool for the extrapolation and interpolation of data to obtain traction values for conditions other than those at which measurements have been made. If relationships between traction characteristics and fundamental properties, relationships between those fundamental properties and molecular structures of fluids, and required traction characteristics for desired lubricant performance are known it should be possible to give guidance to efforts to synthesize lubricant materials for new applications. This guidance would take the form of suggested molecular structures likely to give desirable lubricant performance.

The modelling of these forces is complicated by many factors. In typical lubricated contacts extremely high pressure forces exist, causing the bearing surfaces to undergo significant elastic deflections and causing the viscosity within the fluid to increase by many orders of magnitude. Heating caused by the shearing and compression of the lubricant within the contact counteracts this trend, tending to cause the viscosity to decrease. Measurements from which fluid properties may be determined are very difficult to obtain under the extreme conditions of pressure, temperature, and shear within the lubricated contact. An early attempt to model these forces based on the assumption that the lubricant behaves as a Newtonian fluid was made by Kannell[16] and based on a set of assumptions posed by Crook[17]. Based on the differences in this and similar models with experimental values, it was determined that fluids have both viscous and elastic properties under conditions in the contact. One of the most recent models, formulated by Houpert[18] is based on the assumption of a Maxwell model of rheological behavior of the lubricant posed by Johnson and Tevaarwerk[19]. The challenge and the significance of this problem have resulted in the involvement of numerous investigators too extensive to detail here.

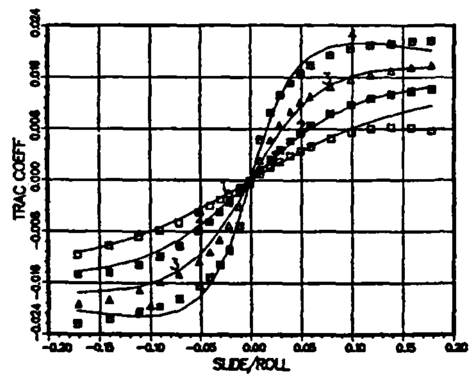
Traction data gathered at the Materials Laboratory has been analyzed using a model formulated by Gupta[20]. This model is similar to the Newtonian model of Kannel discussed earlier and is based on an assumptions of different equations describing the relationship between viscosity, temperature, and pressure within the lubricated contact. The model determines coefficients of the viscosity equation which minimize the squared deviation of the differences between the experimental data and the model predictions. An example of the result is shown in Figure 15, where data for the base stock of an oil used to lubricate electric motor bearings is compared with model predictions for the type I model described by Gupta. The predictions are seen to fit the model predictions, reasonably well, having similar slopes in the linear portion of the traction curves and similar extreme values of the traction coefficient. The temperature change represented in the different plots is seen to have a much larger effect on the traction values than the change rolling speed. The model predictions show a larger effect of load on the tendency toward nonlinearity of the traction curves than is substantiated by the experimental data. The success of traction modelling efforts are greatly affected by the speed, load, and temperature ranges involved and the nature of the lubricating material. Traction models which more accurately represent the behavior of a broad range of materials and which give greater insight into the nature of the lubricating process are being developed.



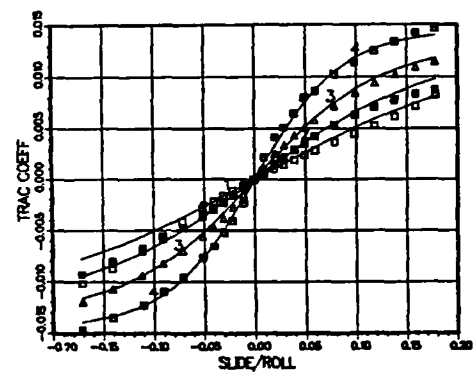
(a) Velocity= 6.9 m/s, Temperature= 25°C



(c) Velocity=10.3 m/s, Temperature= 93°C



(b) Velocity= 6.9 m/s, Temperature= 25°C



(d) Velocity=10.3 m/s, Temperature= 93°C

Figure 15. Traction vs. Slip Speed for Mobil M-28 Base Stock

HYDRAULIC SYSTEM MODELLING

New hydraulic fluids, having more fire resistance and suitable for use over a broader temperature range, are being formulated at the Materials Laboratory. In previous development programs, the technique used to evaluate experimental materials was to test them in an experimental device simulating the mechanical and thermal stresses placed on the fluid by a hydraulic system. Extensive experience with similar materials provided guidance to researchers in ways to improve the performance of materials which failed these tests. Materials now being considered have radically different properties, paiticularly a density nearly twice as great as present materials. To reduce the weight of hydraulic systems using these materials, systems are being developed which use much higher operating pressures than present hydraulic systems. In tests of prototype pumps for these systems, excessive wear and other lubrication related problems have occurred in several components indicating a need for a better understanding of lubrication by hydraulic fluids.

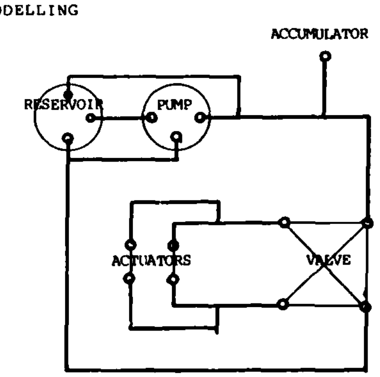


Figure 16. Hydraulic System Model

A schematic depiction of a hydraulic system is shown in Figure 16, which shows pumps, actuators, reservoirs, and lines among the components which make up the system. While the main purpose of the hydraulic fluid is the transmission of mechanical force from the pump to the actuators, a very important secondary function is the lubrication of

bearings and sliding surfaces in pumps, actuators, and other components of the system. The inertia and resistance to flow of fluid within the system adds a dynamic component to the load applied to the pump which may be assessed using models which have been developed for analysis of the system[21,22]. Computer based models exist which address steady state flow and pressure distribution within the system, which conduct thermal analysis to determine the temperature distribution within the system, and which determine frequency response and transient effects of changes in the operating configuration of the system. Each of these has fluid properties[23] among the inputs and the first step of a modelling approach is to conduct studies using these models with the properties of experimental fluids to determine the effects of changes in the fluid properties on the various components of the system. A more detailed approach requires addressing the components on an individual basis.

One type of hydraulic pump is depicted in Figure 17. It is a complex mechanical device, containing many components which are lubricated by the hydraulic fluid. Many of these components have experienced wear and other lubrication related problems in prototype pump tests using experimental components. One of the major components is an angular contact ball bearing, labeled rear bearing, which rotates at the shaft speed and supports the thrust load resulting from the sealing force applied between the cylinder block and the valve plate. Failures of this bearing have been experienced and should be subject to analysis using techniques developed for rolling element bearings discussed earlier in the paper. Radial loads at the other end of the shaft are supported by a roller bearing labeled front bearing. The yoke containing the shoe bearing plate pivots on a pair of roller bearings which are subjected to a thrust load in reaction to the normal and friction forces acting on the shoe bearing plate, which has caused excessive wear in some experimental pumps. These bearings are essentially statically loaded, hence loads may be determined using the static analysis techniques for bearings. Modelling of the wear processes requires more complex techniques, beyond the scope of this paper. Models have been developed[24] with the capability to address the film thickness, pressure distribution and temperature rise at the interface between the piston shoe and the bearing plate, another area where wear problems have been experienced. Many sliding areas, such as the sockets in the piston shoes, where wear has been experienced may be modeled using techniques developed for sliding bearings[25] with suitable geometric adaptations.

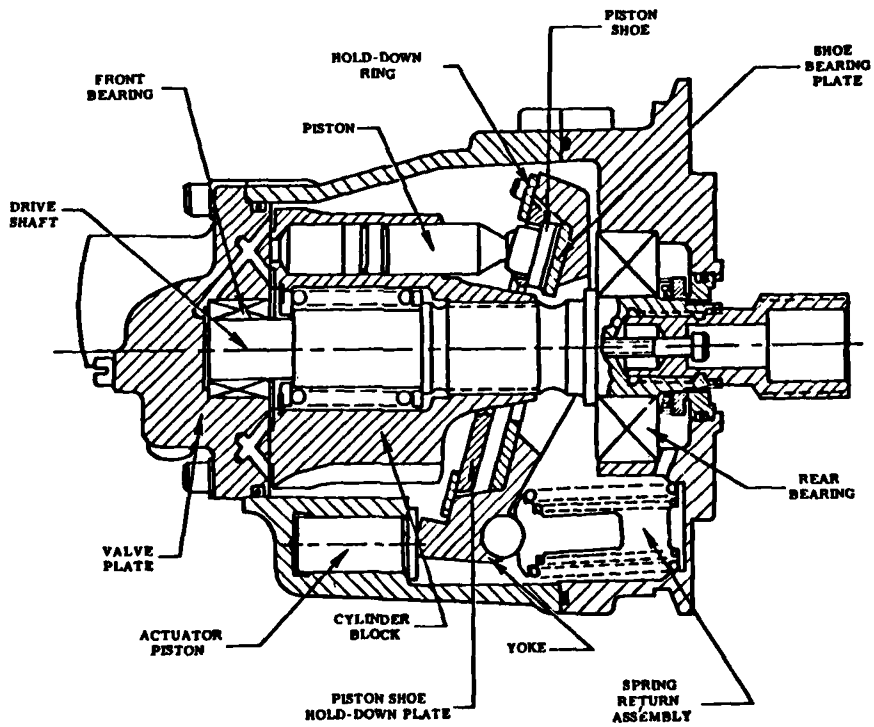


Figure 17. Components of a Hydraulic Pump

It is seen that modelling the lubrication of a single component of a hydraulic system is a complex task involving many different types of models. Models suitable for most of the requirements either exist or can be easily adapted from existing models, but studies need to be conducted using inputs to these models particularized to the systems and fluids being studied. With the results of such studies the functional, lubrication related, requirements on hydraulic fluids can be expressed as physical property requirements. With the requirements so expressed, lubricant synthesis and formulation efforts can be more meaningfully conducted.

CONCLUSION

The application of computerized performance models begins with the formulation of a model describing the response of relevant variables and based on mathematical relationships describing fundamental natural laws or empirical observations. Its successful application also requires validation through correlation of model predictions with experimental results or observations of the behavior of the system simulated, careful formulation of input data and study methodology, and intelligent interpretation of the results of model studies. These techniques have been applied to the modelling of lubricant performance in bearings, to the traction forces arising from the shear of lubricant films, and to the lubrication behavior of hydraulic fluids. The capability of models to simulate the behavior of lubricants under conditions not attainable experimentally, and to simulate the behavior of variables which are not experimentally observable has led to an improved understanding of lubricant behavior which would not be attainable by experimental techniques. The application of modelling techniques can lead to more effective development of lubricant materials by expressing requirements as physical properties, which can be evaluated as soon as the material has been synthesized in sufficient quantities, rather than as functional requirements which can be evaluated only by lengthy, repetitive test programs. It can lead to the more effective application of existing materials to new applications by providing the means to predict the performance of a wider variety of materials in the applications considered.

ACKNOWLEDGEMENTS

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AD-P005 069

SCORING TESTS OF AIRCRAFT TRANSMISSION LUBRICANTS AT HIGH SPEEDS AND HIGH TEMPERATURES

H. Winter and K. Michaelis*

Aircraft engines always contain gears that have to be lubricated under conditions of high speeds and extremely high temperatures. In this field of application scoring damage can be likely to occur. In Europe and partly also in the USA the scoring load capacity of gear oils is expressed in terms of FZG Scoring Load Stage. The FZG Gear Test Rig is described. The normal test procedure A/8.3/90 as standardized in DIN 51 354 using A-type gears at a pitch line velocity of v = 8.3 m/s and a starting oil temperature of 90°C is presented. A modified procedure at double speed and increased oil temperature A/16.6/140 is discussed. The scoring load capacity of aircraft transmission lubricants is worldwide expressed in Ryder Gear Test results. Because of the high costs and problems with the availability of test gears a modified FZG Ryder Test was developed. The method is presented and comparative results of typical aircraft engine oils in the FZG, the FZG-Ryder and the original Ryder Gear Test are shown. From this experience it becomes obvious that alternative test methods for the evaluation of scoring load capacity of aircraft transmission lubricants can be available in the near future.

1. Introduction

With increasing power transmission per volume, increasing speeds and increasing temperatures the demands on lubricants for application in aircraft gearing have increased accordingly. Besides physical properties like viscosity and chemical properties like oxidation stability the technological properties like scoring load capacity have to be tested under conditions close to practice. As scoring is affected by different properties of the lubricant - viscosity as well as chemical composition - gear test procedures were developed and compared with the behavior of the lubricant in practice. These test methods are the Ryder Gear Test in the USA, the IAE test in Great Britain, and the FZG test in Germany. In the following, test procedures on the FZG back-to-back gear test rig and results with aircraft lubricants are described.

2. Standard Test A/8.3/90

The test rig is of the back-to-back type with a mechanical power circuit. The test torque is applied by lever and weights at the load clutch and can easily be calibrated and controlled through the distortion of the torsional shaft (see Fig. 1). A specially designed gear profile, so-called A-type gears with high sliding and thus with high scoring risk are used. Fig. 2 shows the gear profile with the velocity and pressure distribution along the path of contact; table 1 gives the main gear data. Under conditions of dip lubrication at a starting oil temperature of 90°C the gears are run for 15 min. in one load stage (Fig. 3). The load is increased stepwise in 12 load stages until scoring occurs on the flanks. The oil bath is not cooled so that a maximum oil temperature in the twelfth load stage of approx. 140°C is reached. Scoring failure can be assessed either by visual control (Fig. 4) or by weighing the gears after each load stage and plotting the weight loss against transmitted work (Fig. 5). A steep increase in weight loss indicates scoring failure. The scoring load stage is reported and for the gravimetric method the specific wear rate gives an additional value on the relative wear behaviour of the lubricant.

This test method is standardized in Germany in DIN 51 354 /1/ and in Europe in CEC L-07-A-71 /2/. The results of the test express a relative rating of scoring load capacity of different oils. Calculation procedures were established in DIN 3990 and ISO 6336 /3, 4, 5/ for the calculation of the scoring risk of practical gears using the FZG scoring load stage as the "strength value" for the lubricant. Results for typical aircraft oils in the FZG standard test A/8.3/90 are shown in Fig. 6. The relative rating is comparable with results of the Ryder Gear test with these oils.

3. Special Test A/16.6/140

As high performance EP gear oils usually passed the 12 load stages without failure a modified, more severe test procedure at an increased speed of v = 16.6 m/s and for high temperature application at starting oil temperature of 140°C was established /6/. Because the oil is not cooled during the test temperatures up to 180 - 200°C are reached in load stage 12 of this procedure. Again the results of aircraft lubricants are shown in Fig. 7. It is interesting to note the very high scoring load obtained with the Shell 0-160 oil compared with the standard test. The result was proved by duplicate testing.

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According to Maier /7/ a very rapid decrease of the oil performance as related to scoring is observed. This is confirmed with tests on used oils vs new oil in the A/16.6/140 procedure as shown in Fig. 8. Significant reductions in scuffing performance can be seen with increasing use of the oil /7/.

4. FZG Ryder Test R/46.5/74

From different research projects /8, 9, 10, 11/ it is known, that the scoring behavior of different lubricant types with increasing velocity may be very different. In Fig. 9 the scoring load for different base oils and different additive systems is plotted against velocity. From this graph it is evident that a different ranking of the oils is possible when tests are performed at different pitch line velocities.

Because of the high costs of the Ryder test rig and test gears as well as problems with the availability of test gears, a research project was initiated and funded by the German Ministry of Defence. The aim was to develop a test rig and test procedure on a modified FZG test rig which gave equivalent results as compared to Ryder Gear Tests /12/ (Fig. 10).

The principle of the test rig can be taken from Fig. 11. Table 2 lists the differences in the rig design as compared with the original Ryder rig. Because of almost the same center distance of the two rigs FZG R-type gears (Fig. 12) could be designed very close to the original Ryder gears. A comparison of the main gear data is given in Table 3. Table 4 shows a comparison of the gear materials used.

Fig. 13 shows the test procedure. The oil to be tested is heated to the test temperature of 74°C. For every test an oil quantity of approx. 5 l is necessary. The oil is sprayed on the gears at a flow rate of approx. 0.5 l/min. Then the torque of load stage 1 is applied. The load in every load stage can be taken from table 5. The motor is started; the running time is 10 min per load stage. After every load stage the gears are inspected with a microscope and the scored area on each flank is recorded. For this purpose a grid is incorporated in the microscope such that each square represents 5% of the active flank area of a tooth. Thus the location and the size of the scoring marks can easily be recorded. The percentage of damage in each load stage is determined and plotted against the load in a logarithmic chart (Fig. 14). The test is terminated when more than 30% of the active flank is scored. Scoring load is interpolated as the normal tooth load when 22.5% of the flank shows scoring failure. The test on one oil is run with both flanks of the test gear. As result the mean value of all test runs as well as every single result is recorded.

In the course of the investigations a variety of different lubricants - straight mineral oils, mineral based EP oils and different synthetic lubricants - were tested in the FZG Ryder Test and compared to results obtained in the original Ryder machine. The results are plotted in Fig. 15. Fig. 16 shows comparative results for typical aircraft oils in the original Ryder /12/ and the FZG-Ryder test. A very good correlation is observed. In the next few months comparative testing of some commercial oils from different manufacturers and different specification will be done.

The first investigations were started with increased oil temperature to address the increased temperatures in turbines of the future. Tests with 0-160 were performed at 115°C instead of 74°C. The results are shown in Fig. 16. By a modification of the oil spray device even higher temperatures can be achieved when high temperature testing is necessary.

5. Conclusion

Different test methods on modified FZG back-to-back gear test rigs under high speed and high temperature conditions were discussed.

For the evaluation of typical aircraft lubricants a new test procedure FZG R/46.5/74 showed good correlation with test results in Ryder Gear Testing. The costs of the modified FZG test rig and test gears are approximately 25% of a Ryder Gear Test Rig and of original Ryder test gears.

Comparative testing of a wider range of commercial oils will be done in the next couple of months. To meet future requirements of aircraft lubricants in the high temperature range investigations at increased test temperature were started.

6. Acknowledgement

The authors would like to thank the German Ministry of Defence for the funding of the reported investigations and for the possibility to present the results in this publication.

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Center distance	a	91.5	mm
Number of teeth, pinion	z_1	16	-
Number of teeth, gear	z_2	24	-
Module	m	4.5	mm
Tooth width	b	20	mm
Pitch diameter, pinion	d_{w1}	73.2	mm
Pitch diameter, gear	d_{w2}	109.8	mm
Addendum modification, pinion	x_1	0.8635	-
Addendum modification, gear	x_2	-0.5	-
Pressure angle	α	20	deg
	α_{wt}	22.5	deg
Length of recess path	e_1	14.7	mm
Length of approach path	e_2	3.3	mm
Max. sliding velocity	v_{g1}	0.67v *	m/s

Table 1: Geometry of Test Gears Tooth Profile A

* pitch line velocity in m/s

	Original Ryder	FZG-Ryder	Units
load application	axial displacement	torque	
load measurement	recalculated from hydraulic pressure	distortion of calibrated shaft	
center distance	88.9	91.5	mm
pinion speed	10 000	9706	rpm
pitch line velocity	46.5	46.5	m/s
spray lubrication:			
oil flow rate	0.27	0.5	l/min
oil temperature	74	74	°C

Table 2: Comparison of Machines and Operating Conditions of Original Ryder and FZG Ryder Test

		Original Ryder	FZG-Ryder	Units
center distance	a	88.9	91.5	mm
number of teeth	z_1/z_2	28/28	30/30	-
module	m	3.175	3.0	mm
working pressure angle	α_{wt}	22.5	22.5	°
tooth width	b_1/b_2	6.35/26	6.25/26	mm/mm
tip relief gear	C_{a2}	0	15	µm
relative sliding speed	v_{gmax}/v	0.28	0.28	-
material:				
case carburized		AMS 6260	14NiCr14	-
surface hardness	HRC	60-62	60-62	-
surface roughness	CLA	0.3-0.5	0.3-0.5	µm

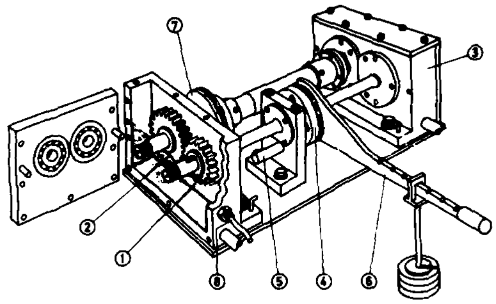
Table 3: Comparison of Original Ryder and FZG Ryder Gears

mean content of	AMS 6260	14 NiCr 14
C	0.11 %	0.15 %
Si	0.27 %	0.25 %
Mn	0.55 %	0.40 %
P_{max}	0.025 %	0.035 %
S_{max}	0.025 %	0.035 %
Cr	1.2 %	0.80 %
Mo	0.12 %	-
Ni	3.25 %	3.5 %

Table 4: Comparison of Gear Materials of Original Ryder and FZG Ryder Gears

Load Stage	Torque in Nm	Tooth Load in N/mm	Tooth Load in lb/in
1	17.5	66	375
2	35.0	131	750
3	52.5	197	1125
4	70.0	263	1500
5	87.5	329	1875
6	105.0	394	1250
7	122.5	460	2625
8	140.0	526	3000
9	157.5	591	3375
10	175.0	657	3750
11	192.5	723	4125
12	210.0	788	4500
13	227.5	854	4875
14	245.0	920	5250
15	262.5	985	5625
16	280.0	1051	6000

Table 5: Load Stages of the FZG Ryder Test



① PRÜFRITZEL
TEST PINION
② PRÜFRAD
TEST WHEEL
③ ÜBERTRAGUNGSGETRIEBE
DRIVE GEAR
④ BELASTUNGSKUPPLUNG
LOAD CLUTCH
⑤ ARRETIERBOLZEN
LOCKING PIN
⑥ BELASTUNGSHEBEL MIT GEWICHTEN
LOAD LEVER AND WEIGHTS
⑦ TORSIONSMEßKUPPLUNG
TORQUE MEASURING CLUTCH
⑧ TEMPERATURFÜHLER
TEMPERATURE SENSOR

Fig. 1: FZG Gear Test Rig (Schematic View)

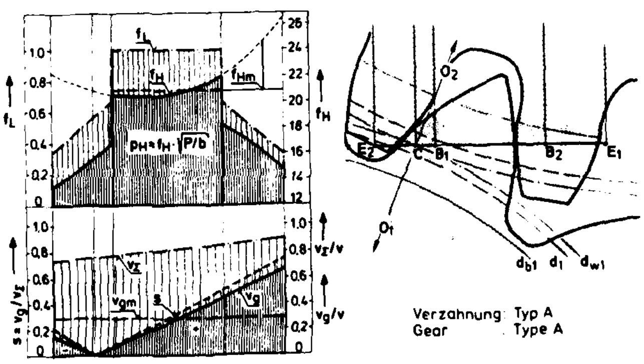


Fig. 2: Load and Speed Distribution (A Type Gears)

GEAR TYPE A

DIP LUBRICATION WITH STARTING OIL TEMPERATURE AT BEGINNING OF EVERY LOAD STAGE ϑ_{OIL} = 90°C, WITHOUT COOLING

PITCH LINE VELOCITY v = 8.3 M/S

DRIVING PINION

LOAD STEPWISE INCREASED UNTIL SCORING OCCURS

FAILURE CRITERION:

VISUAL METHOD: MORE THAN ONE TOOTH WIDTH SCORED AREA

GRAVIMETRIC METHOD: MORE THAN 10 MG OVER THE AVERAGE WEAR RATE

Fig. 3: FZG Scoring Test

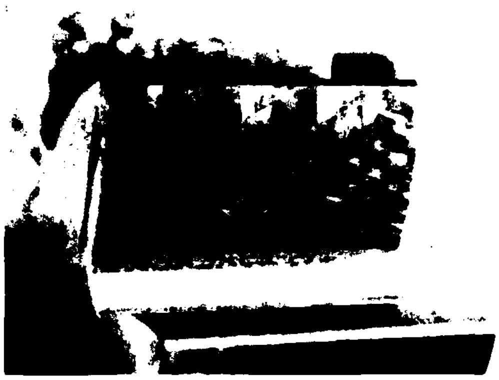
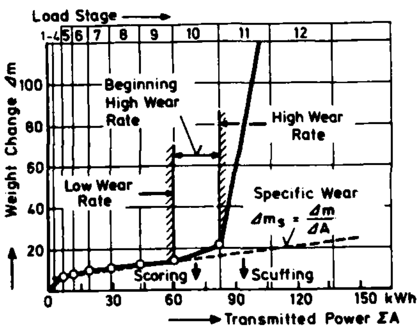


Fig. 4: Flank Appearance
Scoring



Oil Code: Y

Result:
Failure Load Stage 11
acc. Jump in
High Wear Rate

Scoring Torque:
T_{1T} = 450,1 Nm

Specific Wear:
Load Stage 4 to 9

Δm_s below 0,27 mg/kWh
(≈ 0,15 mg/kWh)

Fig. 5: Result of an FZG Test
(Gravimetric Method)

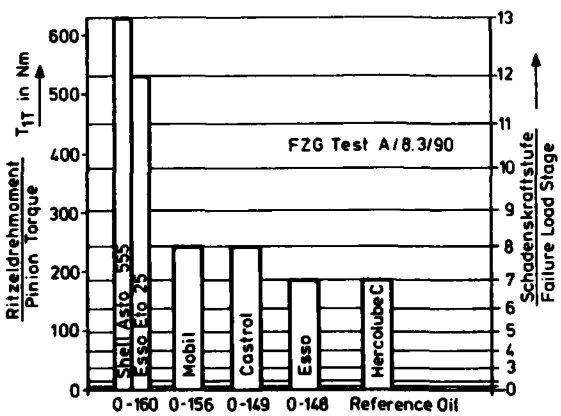


Fig. 6: Scoring Load Capacity
of Aircraft Lubricants in the
FZG Standard Test A/8.3/90

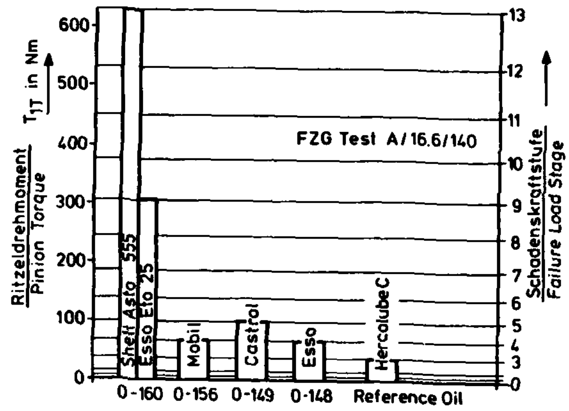


Fig. 7: Scoring Load Capacity of Aircraft Lubricants in the FZG Special Test A/16.6/140

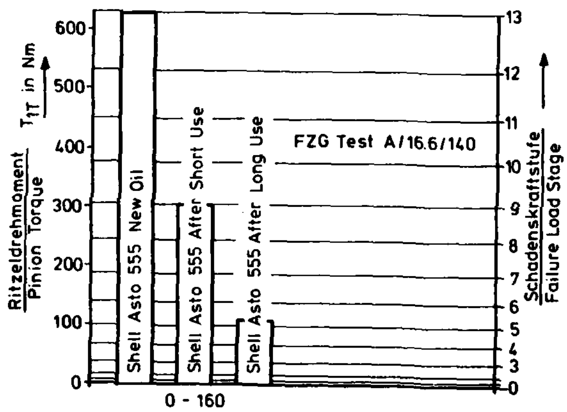


Fig. 8: Decrease of Scoring Load Capacity of Used Oil

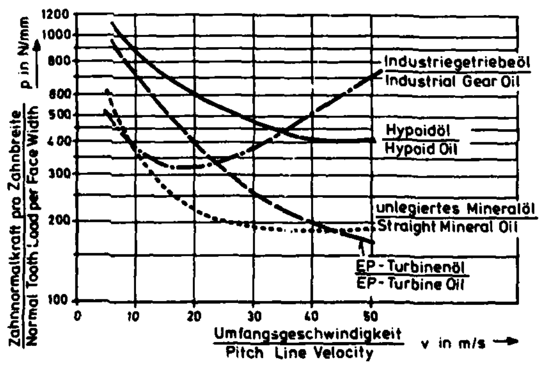


Fig. 9: Scoring Load for Different Gear Oils

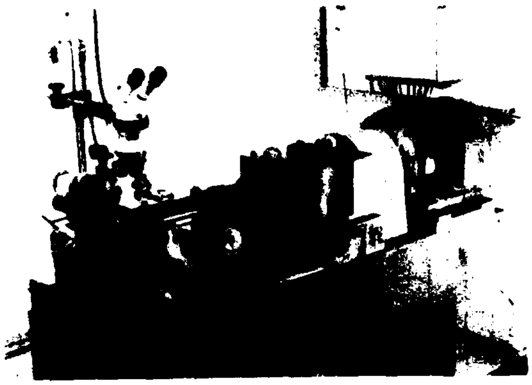


Fig. 10: Photo of the FZG Ryder Gear Test Rig

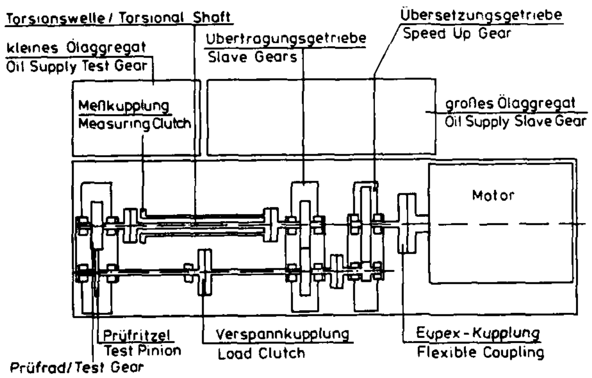


Fig. 11: FZG Ryder Gear Test Rig (Schematic View)

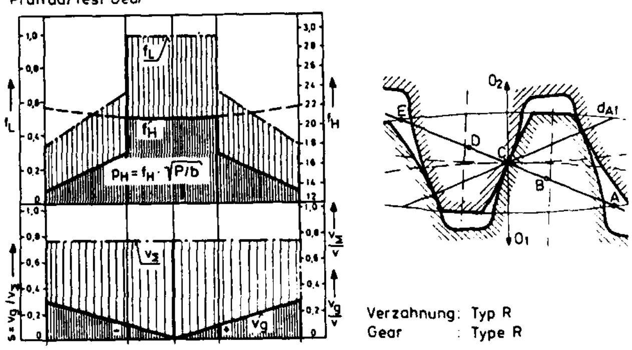


Fig. 12: Load and Speed Distribution (Gear Type R)

GEAR TYPE R

SPRAY LUBRICATION WITH CONSTANT OIL TEMPERATURE OF 74°C, OIL FLOW $\dot{V}$= 0.5 L/MIN.

PITCH LINE VELOCITY v = 46.5 M/S
DURATION PER LOAD STEP T = 10 MIN

LOAD STEPWISE INCREASED UNTIL SCORED AREA ON THE PINION FLANK EXCEEDS APPR. 30% OF ACTIVE FLANK

FAILURE CRITERION:
MORE THAN 22.5% OF ACTIVE FLANK AREA SCORED, SCORING LOAD DETERMINED BY LINEAR INTERPOLATION

Fig. 13: FZG Ryder Test

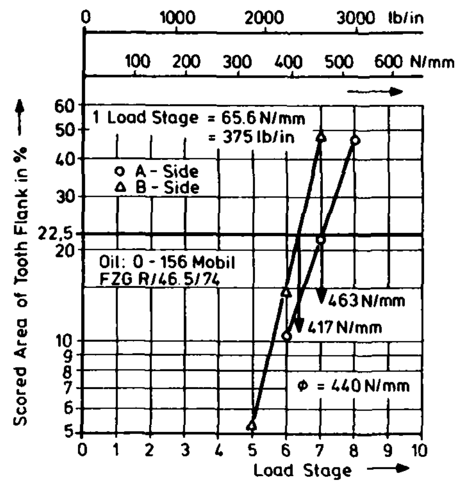


Fig. 14: Evaluation of Scoring Load in the FZG Ryder Test

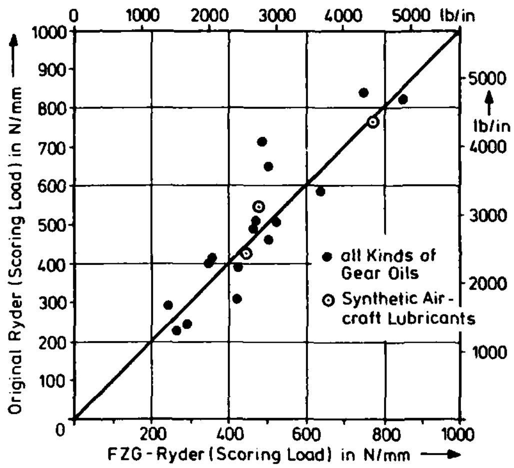


Fig. 15: Comparison of Scoring Load in the Original Ryder and the FZG Ryder Test

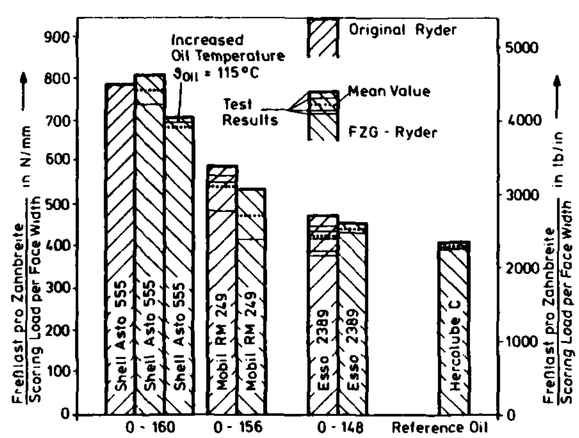


Fig. 16: Comparative Results of Ryder and FZG Ryder Tests for Aircraft Lubricants

AD-P005 070

CRITICAL ANALYSIS OF ACHIEVEMENTS AND MISSING LINKS IN GEAR AND BEARING TRIBOLOGY IN RELATION TO POWER ENVELOPES

by

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SUMMARY

Power envelopes provide a convenient and informative basis of comparison for the development of gear transmissions towards higher power densities. Each such envelope consists of a sequence of segments representing power barriers due to a variety of critical phenomena. Most of these phenomena and barriers are tribologically affected. They may be exemplified by the occurrence of contact between the rubbing surfaces whenever a fully separating lubricant film can no longer be maintained and, further, by pitting and scuffing.

For every gear transmission at least two power envelopes have to be considered in conjunction, i.e. for the most vulnerable pair of gears and rolling-element bearing, respectively. The trends of these two envelopes, as well as their mutual position, depend on the configuration of the major elements in the transmission concerned and on certain tribologic characteristics of the solid rubbing materials and the lubricant.

The power envelopes, being tribologically affected, prove useful for developing tribology into tribotechnology, so as to enable its integration into the design of gear transmissions. Once certain missing links should have been filled in, tribology may thereby provide breakthroughs for developing the technology of both gears and bearings to the high-tech level required for creatively increasing the power densities permissible in gear transmissions.

The present paper concentrates upon those segments of the two power envelopes that are called "contact barriers" in that they relate to the occurrence of contact. In fact, the fundamentals of the other tribologically affected barriers, that of pitting and the one of scuffing, have already been dealt with fairly thoroughly in literature and therefore will here be treated only concisely.

NOMENCLATURE

A	dimensionless group; see definition (4b)
C	non-numerical factor; see formula (3b)
E	Young's modulus
E_r	reduced modulus of elasticity; see definition (2e)
h_{min}	minimum film thickness
$\bar{h}_{min}$	permissible minimum film thickness
k	heat conductivity of the lubricant
p	film pressure
Q	rate of volumetric flow per unit width of band-shaped films
R	radius of conformity; see definition (2c)
T	temperature
T_0	temperature at entry cross-section of the film
V_Σ	sum velocity
$V_{\Sigma,opt}$	optimal sum velocity
W	unit load

α_0 pressure coefficient of viscosity at inlet temperature and ambient pressure

α_0^* representative pressure coefficient of viscosity

η_0 viscosity at inlet temperature and ambient pressure

λ exponent in formula (4a)

Λ specific film thickness; see definition (1)

μ numerical coefficient in formula (4a)

ν Poisson's ratio

ϕ viscous-shear heat per unit time and per unit volume of the lubricant

1. INTRODUCTION

In 1958 the author suggested power envelopes for gear transmissions as a convenient and informative basis of comparison for their development towards higher power densities. The present figure 1 is a refined version of figures 33.1 and 33.2 of that previous paper [Ref. 1], the envelopes being likewise depicted in a log-log-chart by plotting the power transmissible, N, against some such representative speed, n, as that of the smallest gear in the transmission to be considered. The present version could be refined by virtue of recent achievements, mainly those in the theory of elastohydrodynamic lubrication (see section II).

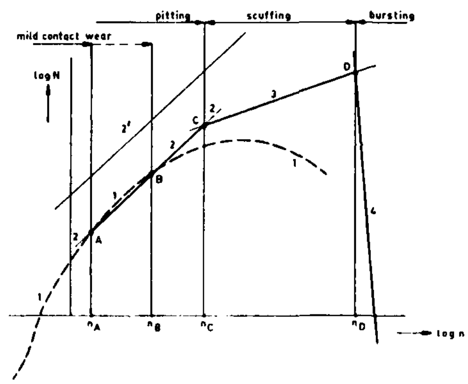

Figure 1 Schematics of Power Envelope, N and n denoting the power transmissible and some representative speed, respectively.

Each such envelope consists of a sequence of segments representing a variety of power barriers that occupy successive speed ranges. Five such barriers have been recognized, that is, those due to tooth breakage, pitting, scuffing, bursting of some gear in bulk by centrifugal overstressing at excessive speed, and finally the occurrence of contact between the rubbing surfaces whenever a fully separating lubricant film can no longer be maintained.

Since, at least for aircraft transmissions, design is such that the tooth breakage barrier is in the present chart about parallel to the pitting barrier (cf. barriers nos. 2' and 2 in Fig. 1) and moreover occupies the higher position, the latter barrier is the decisive one in constituting one of the segments of the power envelope. Further, the bursting barrier, which determines the highest speeds attainable, is absolute in that it cannot be surpassed at all.

Now, in the present paper attention will be focused on the three remaining barriers, nos. 1, 2 and 3, all of which are tribologically affected. Since all of the three corresponding critical phenomena are of a stochastic nature, as will be discussed in the sections to follow, none of the present three barriers is absolute. In fact, the overall position of each depends on the probability deemed permissible by the designer for the critical phenomenon concerned. The fundamentals necessary for assessing the pitting and scuffing barriers, nos. 2 and 3, having already been covered fairly thoroughly in literature, will here be dealt with only concisely (see section 3).

In contrast to the pitting and scuffing barriers, nos. 1 and 2, which are straight or nearly so in the logN-logn-chart of figure 1, the contact barrier, no. 1, shows a curved trend. Of course, since scuffing can take place only during contact, the scuffing barrier must in its entirety lie above the contact barrier. This is clearly brought out in the speed range from n_C to n_D, within which this barrier is to be taken the decisive one (cf. Fig. 1). In this connection it is noteworthy that if by some suitable expedient, such as a more viscous lubricating oil, one should succeed in sufficiently raising the contact barrier, scuffing could be confined to speeds higher than n_C in figure 1.

Since the trends, and in particular the overall positions and slopes, of all three of the present barriers are different, they are bound to intersect each other. More specifically, the particular barrier that at some given speed happens to be the lowest, will be the decisive one in that it determines the point of the power envelope at that same speed. So, when thus exploring the entire speed range, up to bursting, one may construct the entire power envelope from intersecting segments, which in succession represent barriers due to various kinds of critical phenomena.

When speeds are low enough, say below n_A at the point of intersection A in figure 1, the surface distress then occurring on the contact barrier, or even above it, will consist merely in some kind of comparatively mild kind of wear due to sliding under contact. Such wear may well progress so slowly that it may fairly often be deemed permissible in view of the useful life required of the transmission under design.

Further, attention may be drawn to the fact that the contact barrier, no. 1, may well intersect the pitting barrier, no. 2, in two points A and B, located at the speeds n_A and n_B, respectively (cf. Fig. 1). Notwithstanding in the speed range in between n_A and n_B the two rubbing surfaces will not contact each other, pitting may well occur through high film pressures which, after all, may cause surface fatigue just as well as high contact pressures will. In the remaining speed range, extending from n_B up to the transsition at n_C towards the scuffing barrier no. 3, the pitting barrier may also there be considered the decisive one, notwithstanding there it lies higher than the contact barrier. This may readily be recognized by those conversant with the significance of the so-called Λ-ratio, the "specific minimum film thickness", being defined as the ratio between the elastohydrodynamic minimum film thickness calculated and some representative composite height of the asperities on the two rubbing surfaces concerned. In terms of this ratio the two adjacent speed ranges $n_A \le n \le n_B$ and $n_B \le n \le n_C$ relate to the ranges $\Lambda \ge 1$ and $0 < \Lambda \le 1$, respectively.

All in all, perhaps the most useful purpose served by the construction of the above-described power envelopes consists in facilitating the development of the tribology of gears and bearings into their tribotechnology, so as thereafter to integrate the latter in the field of the design and development of gear transmissions, and thus to bring this field to a really "high-tech" level (for an integration generalized for machinery in general, see the author's 1967 paper, Ref. 2). This will be brought out in the following sections which will deal in succession with the three tribologically affected barriers. This will be done by critically analysing the achievements nowadays available for the construction of the power envelopes concerned, whilst pointing out the still missing links. It will then prove that gear designers are well advised to become sufficiently familiar with certain design characteristics that are inherent in lubricating oils as highly essential elements in the rubbing system constituted by the gear transmission to be considered in its entirety. In fact, these design characteristics are just as indispensable as the classical structural ones, i.e. strength and stiffness in bulk of solid elements like the gears and the casing.

Although the sections to follow will be concentrated on the role of the gears in determining a power envelope to the transmission to be considered, the role of the rolling-element bearings will also be discussed, i.e. in the form of still another power envelope.

2. FUNDAMENTALS OF CONTACT BARRIERS

2.1 General Observations

For every pair of meshing gears, and equally for every rolling-element bearing, there is an entire family of contact barriers having the Λ-ratio, that is the specific minimum film thickness, as their parameter. In fact, like with rubbing surfaces in other kinds of moving machine elements, contact is of a stochastic nature bound up with that of the roughness profiles [Ref. 3]. This involves that when the Λ-ratio goes up to about 2 or 3 some contact still occurs, albeit of an increasingly intermittent nature. Accordingly, for instance when the rubbing velocities and the load are kept constant while changing over to a more viscous oil, the kind of "contact wear" then occurring tends to become increasingly milder till there is no such wear at all. Particularly in those rolling-element bearings where skidding is not a problem, and consequently smearing (a kind of scuffing) is neither, even Λ-values a little smaller than unity may prove permissible from the standpoint of useful life.

The aforementioned definition of the Λ-ratio results in the following relationship between the permissible minimum film thickness, $\bar{h}_{min}$, and the chosen Λ-value,

$$\bar{h}_{min} = \Lambda \cdot h_{min} \tag{1}$$

where h_{min} is the more or less fictitious minimum film thickness that can be evaluated from the theory of elastohydrodynamic (EHD) lubrication when conceiving the two rubbing surfaces to be perfectly smooth so that the EHD lubricant film would be impenetrable in the then complete absence of asperities.

So as not to overcrowd figure 1, only one contact barrier has therein been depicted for the application concerned. The various contact barriers of the above mentioned Λ-family will, of course, show different trends and, in particular, different overall positions with respect to those of other kinds of barriers such as those due to pitting and scuffing. In this connection it should be observed that contact barriers are well

conceivable that, in contrast to the one in figure 1, do not show a maximum anywhere in the speed range to be covered.

Furthermore, the designer's latitude in his control over the trends and overall positions of the contact barriers is not confined to his choice of the Λ-ratio. In fact, he has a substantial amount of control over the various quantities that influence the more or less fictitious minimum film thickness, h_{min}, as it may be evaluated from the theory of EHD lubrication. Amongst these influential EHD quantities there are some, like the unit load permissible, from which he may derive the powers that for the various pairs of meshing gears are permissible from the present standpoint at any speed, n, that he takes as representative of the entire transmission. Thus he may find out the particular pair of gears that for the Λ-ratio adopted is the most vulnerable one in that it yields the lowest contact barrier in terms of power transmissible. This barrier is then decisive for the entire transmission, at least in so far as the gears are concerned.

It still remains to establish similar contact barriers for the rolling-element bearings, equally in terms of power transmissible by the gears. Of course, every such bearing should first be rated in terms of its permissible load for the Λ-ratio chosen, and at its individual speed as it corresponds with the speed taken as representative of the entire transmission. Thereafter, through relationship (1) the designer may interpret and depict the performance of every bearing in terms of power transmissible by the most vulnerable pair of gears, and thus find out the particular bearing that is most vulnerable towards contact wear. This can always be done since the configuration of the major elements in the transmission concerned yields not only relationships between the loads on the various gears and bearings but also between their speeds.

Once the most vulnerable bearing is known, its contact barrier can be plotted as one of the segments of the particular power envelope that is representative of the bearings. Thus the designer ends up with two power envelopes, one for the gears and the other for the bearings. Of course, the trends and overall positions of these two envelopes, including their various kinds of barriers as their constituent segments, will in general prove to be different. Finally, the power envelope to be considered decisive for the transmission in its entirety, will have to be constructed from the two sets of segments through selecting those segments that in the successive speed ranges turn out to be the lowest. Thus it is quite conceivable that the decisive envelope consists only partly of the lowest envelope of the gears and for the remainder of one or even more segments of that of the bearings.

To gain more insight into how, primarily in attempts at raising power density, to control power envelopes let us now go into the very basis to contact barriers, that is, into the theory of the elastohydrodynamic (EHD) lubrication. To this end, let us start with the simplest such theory, that for band-shaped and fully flooded EHD films under steady duty where the rubbing velocities are low enough for considering the flow of the lubricant in such films to be isothermal to at least a fair approximation. This will be done in the next sub-section 2.2, whilst the remaining subsections will be devoted to the more complicated EDH theories, such as those necessary for rubbing velocities on the high side, and for EHD films that are elliptical such as is the case of so-called "point contact", or when the duty imposed is not steady but dynamic or transient in that the load imposed, and/or the rubbing velocities, and/or the radii of curvature of the counterformal rubbing surfaces vary with time. In so doing, let us confine ourselves to the evaluation of the minimum film thickness so as later on to deal, if only concisely, with that of the frictional losses generated by the viscous-shear heating in the lubricant film concerned. Furthermore, at least to begin with, the effects of non-Newtonian behaviour, as they may well arise when rubbing velocities are high, will be ignored (for such effects, see item no. 2.4.4 of the present section).

2.2 The isothermal EHD theory for fully flooded band-shaped lubricant films under steady duty

The present EHD theory has been developed by several investigators for cases where, notwithstanding the viscous-shear heating, the lubricant flow generating the pressures in the film may still be considered isothermal. As will be shown in sub-section 2.3, in conjunction with a thermal correction factor the present theory may serve also as a basis to the theory of thermo-EHD lubrication as the latter is to be applied at rubbing velocities higher than those accounted for in the present sub-section.

At least for fully flooded conditions the isothermal EHD theory has long been worked out sufficiently for design purposes. For band-shaped EHD films its results as to minimum film thickness have been conveniently condensed in the form of the "Delft" isothermal EHD chart (Fig. 1 of Ref. 3), which is freely available on application. Let us here confine ourselves to the regime that has been explored most fully in the classical work of D. Dowson and G.R. Higginson [Ref. 5]. After all, this so-called D&H regime covers most applications of counterformal rubbing surfaces with band-shaped films under steady duty and up to moderate rubbing velocities.

For the present purpose and in the D&H regime the relationship between the various influential quantities may, to a fair approximation, be reflected by the following fitted formula, which is a somewhat refined version of the original one of Dowson and Higginson,

$$h_{min} = C.W^{-0.125}.R^{0.425}V_{\Sigma}^{0.70}, \tag{2a}$$

where

$$C = 1.56\ E_r^{-0.025}.(\alpha_0^*)^{0.55}.\eta_0^{0.70}, \tag{2b}$$

the quantities influencing the minimum film thickness, h_{min}, being:
W, the unit load imposed;
R, the radius of conformity, also called the reduced radius of curvature, as defined by,

$$\frac{1}{R} = \frac{1}{R_1} + \frac{1}{R_2}, \tag{2c}$$

where R_1 and R_2 are the radii of curvature of the two cylindrical and parallel rubbing surfaces;

$$V_\Sigma = V_1 + V_2, \tag{2d}$$

the sum velocity where V_1 and V_2 are the two rubbing velocities with respect to the film;

E_r, the reduced modulus of elasticity for plane strain, as defined by,

$$\frac{1}{E_r} = \frac{1-\nu_1^2}{E_1} + \frac{1-\nu_2^2}{E_2} \tag{2e}$$

where E_1 and E_2 denote Young's moduli and ν_1 and ν_2 Poisson's ratios of the two rubbing materials, so that for steel on steel one may put,

$$E_r = \frac{E}{1-\nu^2} = 2.3 \times 10^{11}\ N/m^2 \tag{2f}$$

Further, η_0 stands for the viscosity of the lubricating oil under the conditions in the inlet cross-section of the film, that is, at ambient pressure and at the inlet temperature which may be put equal to the bulk temperature of the gears or the rolling-element bearings, as the case may be.
Finally, α_0^* denotes the so-called "representative" pressure coefficient of viscosity as it is defined in the afore mentioned "Delft" chart [Fig. 1 of Ref. 4], and which for the simple case of an exponential viscosity-pressure relationship just reduces to the coefficient α_0 therein appearing, viz.:

$$\eta(p) = \eta_0 \cdot e^{\alpha_0 \cdot p} \tag{2g}$$

Now, if only for operating conditions where the foregoing findings may be applied, gear designers are well advised to note the following points in so far as an optimization of the position of the contact barrier, or say that of the minimum film thickness h_{min}, is aimed at through some suitable selection of the lubricating oil.

The only influential oil characteristics here to be accounted for are those appearing in factor C of definition (2b), as it is to be substituted into formula (2a). These are the viscosity η_0 and the representative pressure coefficient of viscosity, α_0^*, both relating to ambient pressure and the inlet temperature, or say the bulk temperature of the rubbing bodies bounding the film and contacting it thermally.

In the first place, it is important to note that η_0 and α_0^* appear together in the form of one single "combined oil characteristic", that is, $(\alpha_0^*)^{0.55}.\eta_0^{0.70}$, to which minimum film thickness is directly proportional under the present conditions of nearly isothermal flow in the film.

Further, it is known from viscometric data that, at any given inlet temperature, the range coverable by η_0 with various viscosity grades of any given kind of oils, whether it be mineral or synthetic, is very much wider than that of α_0^*. In any case, with any such given kind of oils the value of α_0^* tends to increase at least somewhat when a higher viscosity grade is chosen [cf. Ref. 4].

Therefore, generally speaking, the choice of a higher viscosity grade will be beneficial from the present point of view. However, there are certain limitations to such a choice.

One of the major limitations relates to difficulties with starting the gear transmission concerned in cold environments. In fact, the higher the viscosity grade chosen, and the higher the corresponding value of α_0^*, the higher the sensitivity of viscosity towards temperature will be for any given kind of oils [again, cf. Ref. 4]. Another limitation consists in the fact that the oil chosen will in general have to lubricate both the gear teeth and the rolling-element bearings. So, with high viscocity grades difficulties with churning losses in the bearings will have to be considered.

Last but not least, with any given individual oil appreciable gains as to the minimum film thickness can be realised through improved cooling, i.e. through thereby increasing the value of the combined oil characteristic $(\alpha_0^*)^{0.55}.\ \eta_0$ by lowering the bulk temperature of the gears and the bearings. This cooling question will be dealt with only under one of the items of section 3, i.e. in connection with the so-called "thermal network method".

Summarizing: For evaluating minimum thickness under the present idealized conditions where, amongst other things, the oil flow in the EHD film concerned may be considered isothermal to at least a fair approximation, designers must consider the combined characteristic of the oil chosen, $(\alpha_0^*)^{0.55}.\ \eta_0^{0.70}$. However, when it comes to higher rubbing velocities, the oil flow may no longer be considered isothermal. In sub-section 2.3 to follow it will be shown that then still another combined oil characteristic has to be introduced, i.e. one in which thermal characteristics of the oil, such as its heat conductivity, enter the picture.

2.3 The thermo-EHD theory for fully flooded band-shaped lubricant films under steady duty

It follows from the empirically known order of magnitude of coefficients of tooth friction that at high speed the heat flux generated by friction on meshing tooth faces may well reach and even exceed 10^6 kW/m^2, which is typical of oxy-acetylene torches. This is true even when an EHD film fully separates such tooth faces, i.e. when the present heat fluxes are wholly generated by viscous-shear heating.

Still another criterion for the intensity of the severe heating concerned is provided by the heat, Φ, generated in some point in the film by viscous shear per unit volume of the oil and per unit time. The average such heat, Φ_{av}, as averaged over the film cross-section to be considered, is simply equal to the ratio between the aforementioned heat flux and the local film thickness. Now, even thicknesses somewhat smaller than 1 µm are still representative of fully separating EHD films on meshing gear teeth and in rolling-element bearings. Further, it has long been recognized that, especially at high rubbing velocities, the viscous-shear heat Φ is far from uniform across the cross-section of the film concerned [Ref. 6]. So, the viscous-shear heating may well become so intense that it may be compared with the situation where one litre of the oil is continuously exposed to the extreme heat that would be released by the total conversion of all the electric power producible in a highly industrialized country. Clearly, if such intense heat should not be withdrawn continuously in a most effective manner, nothing much would remain of the oil.

Now, as is evidenced by the very existence of fully separating films, even at rather excessive rubbing velocities, the temperatures in the film, and including these on the two rubbing surfaces, will not usually reach the level where the oil would disappear by evaporation and thus cause the film to collapse by "implosion". This can be explained on the basis of two facts. First, notwithstanding oils are poor heat conductors, the films concerned are so thin that even the present intense viscous-shear heat can still be conducted away fairly efficiently towards the two rubbing bodies [Ref. 6]. Second, the EHD film is in effect a moving heat source to which, at the high rubbing velocities concerned, the rubbing surfaces are exposed only during rather short time intervals. For further particulars on the withdrawal of frictional heat from moving heat sources the reader may consult the flash temperature theory, developed already in 1937 (for a later, more accessible survey and refinement of this theory, see Ref. 7).

A discussion on both thermal and mechanical stability of the EHD films will be given only in sub-section 2.4. In any case it will now be clear that at high rubbing velocities the isothermal EHD theory of sub-section 2.2 must be replaced by a thermo-elastohydrodynamic one in which the thermal effects of the viscous-shear heating are well accounted for through thermal boundary conditions that are sufficiently realistic in the light of the flash temperature theory.

From the work of several investigators it may be concluded that the most convenient thermo-EHD design calculation consists in applying a "thermal correction factor", C_{th}, to the isothermal minimum film thickness. That is, first determine h_{min} as it is found by substituting into the isothermal EHD formula (2a, b) the mechanically influential quantities appearing therein, and then multiply it by the factor C_{th} which has still to be expressed in terms of certain thermally influential quantities.

For quite a variety of operating conditions W.R.D. Wilson and his associates have set up empirical formulae fitted to their thermo-EHD evaluation in a series of papers, starting in 1975 [Ref. 8] and ending up in 1983 [Ref. 9]. A certain approximation is involved because of some simplifying and more or less tacit assumptions of which two major ones are:

1. the rate of flow per unit width of the band-shaped films, Q, would be determined solely in their inlet zones, these acting as a boosting, or rather priming, pump to the remainder of the film concerned;

2. although Wilson et al. have well accounted for the thermal effects induced by the viscous-shear heating in conjunction with the variation of viscosity with both pressure and temperature, they have, in contradiction with the aforementioned flash temperature theory, assumed the distributions of the temperatures on the two rubbing surfaces to be uniform, if only along the inlet zone.

In this connection it is first worth noting that the rate of flow Q may be conveniently expressed in terms of the sum velocity, V_Σ, and the so-called "central film thickness" h_c, which is representative of the nearly parallel conjunctional zone of typical EHD films, viz.:

$$h_c = \frac{2Q}{V_\Sigma} \tag{3a}$$

Further, from the well-established fact that the ratio h_{min}/h_c rarely falls outside the following comparatively narrow range, and if so only to a negligible extent,

$$0.7 \le \frac{h_{min}}{h_c} \le 0.8, \tag{3b}$$

it may be concluded that the thermal correction factor C_{th}, as originally relating only to Q or say h_c, may to a fair approximation be applied also to the isothermal EHD minimum film thickness of formula (2a, b).

To exemplify certain inaccuracies in Wilson et al.'s results, and the consequent need of refinements, let us confine ourselves to thermo-EHD lubrication under conditions of pure rolling. For these conditions their fitted formula reads,

$$C_{th} = [1 + \mu . A^{\lambda}]^{-1}, \tag{4a}$$

where the dimensionless group A is defined by:

$$A = \frac{\beta_0 \eta_0 V_\Sigma^2}{k}, \tag{4b}$$

k standing for the heat conductivity of the oil concerned, and β_0 for the absolute value of the temperature coefficient of viscosity under ambient pressure and at the inlet temperature T_0,

$$\beta_0 = -\frac{1}{\eta_0} \cdot \left(\frac{d\eta}{dT}\right)_{T=T_0}, \tag{4c}$$

so that $\beta_0\ \eta_0$ equals the opposite to the slope, $(d\eta/dT)_{T=T_0}$, of the vicosity-temperature curve for ambient pressure.

For the coefficient λ and the exponent μ Wilson et al. found for the fitted formula (4a),

$$\lambda = 0.108 \text{ and } \mu = 0.62 \tag{4d}$$

On the other hand, refraining from the simplifications inherent in Wilson et al.'s aforementioned assumptions nos. 1 and 2, Ghosh and Hamrock in their recent paper [Ref. 10] could fit their more accurate results by means of the following values (see their formula (36) and Tab. 4),

$$\lambda = 0.0574 \text{ and } \mu = 0.40. \tag{4e}$$

One conclusion is that Wilson et al. underestimated the thermal correction factor C_{th} (see also Tab. 4 of Ref. 10), thus remaining on the conservative side in so far design calculations of the contact barriers for pure rolling are concerned.

Further, the discrepancies between the two pieces of work have presumably to be attributed mainly to the simplification inherent in Wilson et al.'s assumption no. 1. In fact, as regards their assumption no. 2 and at least in the range of rubbing (rolling) velocities covered, Ghosh and Hamrock found negligibly small non-uniformities of the distribution of the surface temperatures along the inlet zone (this is evidenced by the small temperature rises of the solid surfaces, of less then 1 °K at the end of the inlet zones under the pure rolling conditions in figure 8 of reference 10).

However, it is felt that at rubbing velocities higher than so far covered, and at which refinements of the thermo-EHD theory will be more valuable, the non-uniformity of the surface temperatures in the inlet zone will no longer be negligible. So, thermal inlet zone analyses along the lines of Wilson et al., i.e. those retaining assumption no. 1, would then even for pure rolling call for refinements by dropping their assumption no. 2. That is, the effect of the present non-uniformity on the rate of flow, or say on the central and the minimum film thickness, would then have to be accounted for, like in Ghosh and Hamrock's analysis, by means of thermal boundary conditions based on the flash temperature theory.

Further, Ghosh and Hamrock's contention that sliding would have negligible effect on the film thickness, especially at low rolling velocities, may well prove untenable with sliding as fast as on the meshing tooth faces in high-speed gears. Then, also in thermal inlet zone analysis would refinements of the aforementioned kinds be necessary.

Finally, let us go into the question what characteristics of both the lubricating oil and the solid rubbing materials are influential in thermo-EHD lubrication, and in particular how these can be exploited for design optimizations of the minimum film thickness, or say of the trend and overall position of the contact barrier.

For the details of such an optimization in high-speed roller bearings, in particular in so far as a suitable selection of the lubricating oil is concerned, the reader is referred to the author's 1978 paper [Ref. 4]. Let it here suffice to state that in such bearings the minimum thickness of the thermo-EHD films on both the inner and outer raceways was found to go through a maximum when the sum velocity reaches an optimum, $V_{\Sigma,opt}$, defined by:

$$V_{\Sigma,opt} = 7.42\left(\frac{k}{\beta_0\ \eta_0}\right)^{1/2}, \tag{5}$$

which is a simple function solely of a "combined non-isothermal characteristic" of the oil selected, $k/\beta_0\ \eta_0$. It is important to note for design purposes that this characteristic, which has already come to the fore in the thermal correction factor C_{th} and in the dimensionless group A (see formulae (4a and b)), is dependent only on the kind of oil selected and on the inlet temperature of the thermo-EHD film.

True, the present expression (5) for the optimal sum velocity was based on the somewhat approximative thermal inlet zone analysis of Wilson et al. (see formulae (4a) and (4b) in conjunction). But a revision by means of the more refined analysis of Ghosh and Hamrock (see formulae (4a) and (4e) in conjunction) yields an expression for $V_{\Sigma,opt}$ that does not differ from the former (5) to any spectacular extent. Therefore either analysis will suffice for first estimates for design purposes, such as the following ones.

By means of formula (5) the author demonstrated in his 1978 paper [Ref. 4] that the lubricating oils specified by roller bearing manufacturers show values of their combined non-isothermal characteristic, $\beta_0\ \eta_0/k$, such that the DN-values corresponding with the optimal sum velocities, $V_{\Sigma,opt}$, are rather high but

still distinctly lower than the highest DN-values reached nowadays. So, at "super-optimal" sum velocities or DN-values, the present oil characteristic, in conjunction with the "combined isothermal oil characteristic" $(\alpha_0^*)^{0.55}.\eta_0^{0.70}$ (see sub-section 2.2, and in particular formula (2a, b)), provides an important tool in selecting oils for optimizing the minimum thickness of thermo-EHD films.

Further, by using formula (2a, b) in conjunction with formulae (4a, b) and (4d), it can be shown that even up to at least somewhat beyond the optimal DN-values, mineral oils with low viscosity indices will still be beneficial for increasing the minimum EHD thicknesses in roller bearings. In fact, then the detrimental effect of the corresponding increase of the temperature coefficient of viscosity β_0, and thus that of the decrease of the thermal correction factor C_{th} (see formulae (4a, b and d)), will be more than offset by the beneficial effect of the concurrent increase of the representative pressure coefficient α_0^* in the combined isothermal oil characteristic $(\alpha_0^*)^{0.55}.\eta_0^{0.70}$ (see formula (2a, b)). However, since for non-mineral, synthetic oils their various correlations between α_0^* and β_0 differ from the one for mineral oils, the present finding will not in general be valid also for the former kinds of oils.

Finally, the slopes, $\beta_0\ \eta_0$, of the viscosity-temperature curves of different oils at different inlet temperatures show a range much wider than that of the heat conductivities. So, nothing much can be achieved in optimizing thermo-EHD minimum film thicknesses through selecting oils for their heat conductivity.

It still remains to be seen what kind of thermal characteristic(s) of the solid rubbing materials influence the minimum thicknesses of thermo-EHD films, and to what extent they may be exploited for optimizing these thicknesses. Now, it follows from the flash temperature theory that, practically speaking, the so-called "thermal contact coefficient" is the only such thermal characteristic. It is defined as the square root of the product of the heat conductivity and the specific heat per unit volume of the solid material concerned.

As has already been pointed out in the discussion following formulae (4d) and (4e), one may, at least for operating regimes with pure rolling at not too high velocities, use the comparatively simple thermal boundary condition that imposes uniformity of the surface temperatures, albeit not beyond the end of the thermal inlet zone of the thermo-EHD film concerned [Refs. 8 and 9]. However, this boundary condition is not compatible with the flash temperature theory. Accordingly, the effects of the thermal contact coefficient concerned cannot possibly have reflected themselves in the results of Wilson et al.'s thermal inlet zone analysis [Refs. 8 and 9]. Remarkably enough, these effects are neither revealed, at least not clearly so, in the results of Ghosh and Hamrock's analysis, which is more refined in that the flash temperature theory has been well accounted for. All in all, it would appear that, if not beyond the above-defined pure rolling regime, the loss in accuracy of the thermo-EHD minimum film thicknesses is at least bearable for quick design estimates.

However, it is to be expected that in the regime of higher rubbing velocities, and certainly whenever fast sliding occurs, Wilson et al.'s thermal boundary condition will have to be replaced by that of Ghosh and Hamrock. Accordingly, in this regime the thermal contact coefficients of the solid rubbing materials must become distinctly influential. Further, in this connection it may be noted in passing that, certainly in the operating regime with which we are here concerned, the thermal contact coefficients may well become much more influential to the frictional losses induced in the thermo-EHD films, or say as to the coefficients of friction or traction. The reason is that in this regime these losses are primarily induced in the conjunctional film zone where, in accordance with the flash temperature theory, the viscosity of the oil is highly affected by the comparatively high temperatures in that same zone.

Finally it should be considered that the range of thermal contact coefficients covered by the various hard steels, which ordinarily are indispensable for counterformal rubbing surfaces under heavy duty, happens to be rather narrow. So, nothing much can be achieved as to optimizing thermo-EHD minimum film thicknesses through changing over from one such steel to another. Furthermore, again for the same purpose, a change-over from steels to ceramics, for instance in rolling-element bearings and bearing in mind the much lower thermal contact coefficients of the latter materials, may well prove at least somewhat detrimental at the high rubbing velocities in the present regime.

Summarizing the present sub-section 2.3: The foundation presented so far, even when broadened by the refinements suggested in the foregoing for filling in certain missing links, is still confined to the comparatively elementary category of thermo-EHD lubricant films that is subject to the following five restrictive features: 1) band-shaped films; 2) full flooding of the films; 3) films under steady duty only; 4) films remaining both mechanically and thermally stable so that they do not collapse; 5) throughout the films the flow behaviour of the lubricant may still be considered Newtonian. Since designers will in general have to cope with less restrictive categories of thermo-EHD films, sub-section 2.4 may serve as a survey of the questions involved when having to deal with one or more of the present five features.

2.4 Survey of more general categories of thermo-EHD lubricant films

The present survey is subdivided into items nos. 2.4.1 - 2.4.5, which will deal consecutively with the above-stated five features of which one or more are characteristic of categories of thermo-EHD films more general than those of the band-shaped films covered by the foregoing sub-sections 2.1, 2.2 and 2.3. These items are: no. 1, "Elliptical EHD Films"; no. 2, Effects of Starvation"; no. 3, Mechanical and Thermal Instability"; no. 4, "Dynamic Duty and the Quasi-Steady Approach"; no. 5, "Accounting for Non-Newtonian Flow Behaviour".

2.4.1 Elliptical EHD films

In the numerous kinds of gears and rolling-element bearings having geometries resulting in so-called "point contact", the EHD films are not band-shaped but elliptical. In fact, the conjunctional zone of such films, which by definition coincides with the contact area that then in the absence of the lubricant film would be elliptical in accordance with Hertz's theory of elastic contact under the given load.

In Part II of their recent paper [Ref. 11] Chittenden et al. have succeeded in well rounding off the isothermal EHD theory for the kinematically most general and frequently occurring cases that may arise with elliptical films. These are the cases where either rubbing velocity shows a direction at an arbitrary angle to the major or minor axis of the films concerned. This generality is the more important since, at least with gears having "point contact", the directions concerned will vary with time during any given meshing cycle.

Chittenden at al.'s theory has been worked out only for fully flooded elliptical films under steady duty. Nevertheless it provides a valuable basis of comparison with the more severe applications where the rubbing velocities are high enough to call for refinements and extensions for, among other things, the effects of non-isothermal oil flow in the film and those of starvation on the then thermo-EHD films.

2.4.2 Effects of starvation

Most of the work so far available for EHD films, both band-shaped and elliptical ones, has been done for so-called "fully flooded" conditions. These conditions are usually defined on the basis of models in which the theoretically adapted contours of the two rubbing surfaces are assumed to extend to infinity, and where the oil is, again theoretically, supplied in an entry cross-section lying at an infinite distance upstream the conjunction zone of the EHD film concerned. With such models it was found that the rate of flow thus generated through the entire film in this theoretical manner, induced the following effect: The higher this rate of flow the higher the so-called central film thickness h_c; accordingly, at the given sum velocity V_Σ (for band-shaped films, see formula (3a)), the higher the minimum film thickness h_{min} (again for band-shaped films, see range (3b)). Now, in actual practice one can usually ensure a sufficiently abundant supply of oil so as to leave it to the boosting section in the inlet zone to generate the rate of flow required for the EHD film as a whole. But, of course, designers cannot achieve this by a supply at infinity.

Fortunately, by retaining the aforementioned models with contours that still extend to infinity, but putting the entry cross-section at various finite distances upstream the conjunctional zone, one may derive the corresponding rates of flow, as well as the central and minimum film thicknesses, as a function of these distances. Conversely, once the designer specifies the minimum film thickness h_{min} to be aimed at, he may estimate the corresponding central film thickness from range (3b), and finally specify the rate of flow required (for band-shaped films, see formula (3a)). It is admitted that the accuracy of such estimates is no better than that involved by range (3b).

However, when it comes to specifying the minimum film thickness, $\bar{h}_{min}$, permissible from the standpoint of the contact barrier aimed at, the designer will first have to specify the Λ-ratio (see definition (1)). Now, in so doing he must first prescribe some specification of the roughness of the two rubbing surfaces. But the usual roughness standards incorporate tolerances, or say uncertainties in the design stage, of at something like ± 25 per cent. So, estimates based on range (3b), which incorporates uncertainties of only about ± 7 per cent, will suffice for all practical design purposes.

As a first question regarding the prevention of undue starvation it remains to ensure that in operation the rate of flow thus specified will actually be generated, that is, mainly by the boosting action taking place in the inlet zone of finite extent.

Now, in theoretical literature on starvation and its effects on minimum thickness of EHD films, these effects are expressed as a function of the extent of the inlet zone, i.e. of the finite distance between the entry cross-section and the leading edge of the conjunctional zone [Refs. 12 and 13 for band-shaped and elliptical EHD films, respectively]. However, the results thus obtained in terms of this function cannot be used by designers in any direct manner. The reason is that they cannot very well prescribe the aforementioned distance, or say, extent of the inlet zone. So, it still remains to recast the theoretical results so far available on starvation into a form which contains the rate of flow instead of the present extent.

A second question emerges from the well-known possibility that even when the supply of the oil in bulk to the moving surfaces in full motion is super-abundant, starvation may yet occur, e.g. owing to centrifugal fling-off from gear teeth before entering mesh and/or to interference by the tips of incoming teeth which results in cut-off of oil supply jets. So, this question too must be considered, for instance through the data and insight evolved in fairly recent papers [Refs. 14, 15 and 16].

Finally, a third question arises from the fact that the second becomes most important especially at high rotational speeds, where moreover the rubbing velocities also tend to be high. So, there is a definite need to extend the investigations so far available only for starvation in isothermal EHD lubrication to thermo-EHD lubrication.

2.4.3 Stability of EHD films

In the aforementioned theories of isothermal EHD and thermo-EHD lubrication it is tacitly assumed that the films therefrom evaluated would be stable. However, as long as these films have not theoretically and/or experimentally been proven to be stable indeed, the theoretical results, such as those about minimum film thicknesses, cannot be relied upon. Now, at least the following three kinds of instability will have to be examined, of course including the delimitations between the corresponding regimes of stability and those of instability.

The simplest kind of instability is the purely mechanical one, which is the one relating primarily to isothermal EHD films, that is, the films in which at the comparatively low rubbing velocities here implied the oil flow may be assumed isothermal to a fair approximation. In his 1984 paper [Ref. 17] M.M. Kostreva has claimed that such a mechanical instability is bound to occur in a certain regime of isothermal EHD lubrication, and at that under steady duty.

However, if the delimitation of Kostreva is plotted in the aforementioned "Delft" isothermal EHD chart (Fig. 1 of Ref. 4), it is seen to fall in the Dowson and Higginson regime to a quite substantial extent. Now, there are many applications of which the latter regime is representative whereas, to the best of the author's knowledge, neither in actual practice nor in experiments has the present kind of instability been conspicuous, at least not at the comparatively low rubbing velocities with which we are here concerned.

So, it would appear that, perhaps owing to some oversight, Kostreva's theory is not sufficiently representative. Possibly a kind of inverse treatment might prove both easier and more fruitful. This is a treatment in which the analysis for instability is not based on a perturbation technique applied to the film profile but on one relating to the distribution of the pressures generated in the film.

Two more kinds of instability are thermal, or rather thermo-mechanical ones. They might occur at rather high rubbing velocities, that is whenever the withdrawal of the viscous-shear heat by the two rubbing bodies should become inadequate. If occurring, they would result in a collapse of the film much more disastrous than that due to mechanical instability. In fact, at the high rubbing velocities involved, and especially when sliding is also fast, the present collapse would readily develop into scuffing.

The simplest kind of thermal instability is the one where one might still ignore the effects of the thermal distortion of the two rubbing surfaces, and thereby of the thermo-EHD film profile, as it is induced by the non-uniform temperature fields inside the two rubbing bodies and as it has been studied by R.A. Burton [Ref. 18].

The other, even more complex kind of thermal instability is the one where, probably only at rather excessive rubbing velocities, the aforementioned thermal distortion can no longer be ignored.

In tackling the problems involved in these two kinds of thermal instability the findings of Dakshina Murthy [Ref. 6] about certain limitations to non-isothermal flow in lubricant films may well prove helpful. Further, those interested may be referred also to some of the papers published in the proceedings of three workshops held on thermomechanical effects in sliding systems [Refs, 19, 20 and 21].

2.4.4 Dynamic duty and the quasi-steady approach

Even in gear transmissions that in bulk run under steady duty the EHD films between the meshing tooth faces, as well as those in the rolling-element bearings, are subjected to dynamic duty. For instance, in straight involute spur gears, where these films are band-shaped, three of the various elastohydrodynamically influential quantities in Dowson and Higginson's formula (2a, b) will vary with time during every meshing cycle, i.e. from one meshing position to another. These quantities are, the unit tooth load W, the radius of conformity R, and the sum velocity V_Σ. The variation of the latter two quantities can readily be determined from the well-known geometry and kinematics of involute toothing systems. But the variation of the load is much more complex, and at that bound up with unavoidable uncertainties in the design stage. In fact, this variation is caused by the inertial effects induced not only by the calculable deflections of the meshing teeth but also by the toothing errors, the latter being afflicted with the uncertainties due to the tolerances to be granted the workshop. On the other hand, in roller bearings, provided skidding may be ignored, the situation is much simpler. In fact, both the radius of conformity, R, and the sum velocity, V_Σ, there remain constant whilst only the unit load will vary during the "planetary" cycle of each roller.

Now, as has been pointed out in the foregoing, the EHD theory has, notwithstanding certain missing links, already been worked out fairly satisfactorily when it comes to design calculations for steady duty. However, judging from even the most informative papers on the theory of EHD lubrication under dynamic duty, much still remains to be done before designers can sufficiently rely upon this theory. For instance, in his 1971 paper J.P. Vichard [Ref. 22] has gone no further than evaluating models in which for the band-shaped EHD films concerned only one of the three above-mentioned influential quantities was allowed to vary at a time. So, for non-skidding roller bearings his EHD theory may well provide a basis for evaluating the variation of minimum film thickness with that of unit load throughout the planetary path of each roller, and on both the inner and outer raceways. However, this theory, having been developed only for isothermal EHD lubrication is not suitable for high-speed bearings.

For the more complex problems with meshing tooth faces, where all three of the aforementioned elastohydrodynamically influential quantities vary simultaneously, no satisfactory EHD theory, not even an isothermal one, seems to be available yet. Especially when encountering such problems, designers tend to fall back upon the so-called "quasi-steady approach". Because of its simplicity this approach is the most attractive one for estimating EHD minimum film thicknesses under dynamic duty. But it would appear that so far nothing much has been published as to the delimitation of its range of applicability. This situation urged the author to tackle this problem. He has meanwhile succeeded in estimating this delimitation in a fairly simple manner. But because of the space required he prefers to postpone its publication to a later occasion. Only the major thoughts behind this estimate will be adumbrated herebelow.

First, attention is focused on the fact that the quasi-steady approach implies all of the successive instantaneous film profiles to be identical with those following from Dowson and Higginson's isothermal EHD theory for steady duty. In turn, this implies that, to the approximation concerned, at every instant the squeezing velocities, these of normal approach of the two rubbing surfaces, are known over the entire extent of these surfaces. Further, this extent includes that of the nearly parallel conjunctional zone of every instantaneous EHD film profile and also that of the two "wings" on either side, So, the instantaneous distribution of squeezing velocities must be fairly uniform over the conjunctional zone but distinctly non-

uniform over either "wing". Now, the latter, non-uniform sqeezing velocities are reminiscent of the flapping exercised by birds in flight. Therefore, the corresponding instantaneous contribution to the elastohydrodynamic generation of the oil pressures in the film may appropriately be called the "flapping" action (for the effects of this action in journal bearings under dynamic duty, see Ref. 23).

Now, especially in certain cases of severe dynamic duty the flapping action, in cooperation with the more or less uniform squeezing action in the conjunctional zone, may well become so appreciable as to result in increments to the EHD film pressures following from Dowson and Higginson's theory. For instance, when the two combined actions are such that these increments are positive, they will cause the conjunctional portion of the film profile to assume the form of a crater. Accordingly, the delimitation sought for the range of applicability of the quasi-steady approach can be estimated on the basis of the condition that the aforementioned pressure increments remain small enough for not violating the near-parallelism of the conjunctional film portion which is considered typical of Dowson and Higginson's steady-duty regime.

Let it further suffice to point out that the range of applicability thus estimated for the quasi-steady approach, proved to be wider than the author had surmised previously.

Finally, it is remarkable that in literature on EHD lubrication under dynamic duty several models have been simplified to the extent that they cannot allow for the cratering phenomenon. So, such models are bound to overlook the promotion of squeeze retardation through partial entrapment of the oil in the conjunctional zone, as it will occur with cratering (for such squeeze retardation in compliantly lined journal bearings under dynamic duty, see Ref. 24).

2.4.5 Accounting for non-Newtonian flow behaviour

The aforementioned EHD theories relate only to cases where, under the duty to be considered, the flow behaviour of the lubricant may be assumed to be Newtonian throughout the lubricant film. So, wherever the flow behaviour, or say the rheologic characterization, had to be accounted for, the sole knowledge of the viscosity-temperature-pressure-relationship of the lubricant selected was taken to suffice.

However, as has long been recognized from the work of many investigators, at high rubbing velocities the lubricant will be subjected to rates of shear so high that it may no longer be considered Newtonian, at least not in some sizable portion of the EHD film. Fortunately, it would appear that even up to rather high velocities at least the greater part of the inlet zone still falls outside that non-Newtonian portion (cf. Fig. 24 of Ref. 24). So, in view of the aforementioned dominance of the boosting function of the inlet zone in determining the rate of flow through the entire film, the "Newtonian" minimum film thicknesses still tend to be fairly reliable as design estimates. Even so, there is still a need for work on delimiting the range of presumably rather excessive rubbing velocities where the effects of non-Newtonian behaviour may no longer be ignored when it comes to predicting minimum film thicknesses. In this connection it may finally be noted that, except in pure rolling, the frictional losses will, in contrast to the rates of flow, be generated mainly in the conjunctional film zone. Since non-Newtonian behaviour will in general reveal itself first of all in this zone, the "Newtonian" coefficients of friction or traction may certainly not be relied upon at the high rubbing velocities concerned.

Now, it has taken a long time to unravel non-Newtonian behaviour as it is typical of lubricants under EHD conditions. Since the number of non-Newtonian rheologic characteristics exceeds that of the Newtonian ones, the "non-Newtonian" EHD theories will be even more complex than the more or less classical "Newtonian" ones. This may explain why there is still a paucity of at least a sufficiently broad survey of evaluated "non-Newtonian" solutions.

For an adequate characterization of the non-Newtonian behaviour of lubricants in EHD films the reader may for shortness be referred to the survey given by W.O. Winer in his 1983 paper [Ref. 25] about his and his associates' work. Let it here suffice to point out that according to this work the relationship between shear stress and shear rate will at a certain transitional state change over from the simple liquid-like, Newtonian viscous behaviour to the non-Newtonian one typical of an amorphous, elastoplastic and more or less glassy solid.

3. FUNDAMENTALS OF THE PITTING AND SCUFFING BARRIERS

Already in the beginning of the century pitting, or say surface fatigue, was recognized, first of all for rolling-element bearings, as one of the tribologically affected critical phenomena limiting the useful life of counterformal rubbing surfaces. Ratings and design calculations then established for such bearings in terms of load capacity as a function of the number of fatigue cycles, have since been based primarily on Hertz's classical theory of elastic contact, i.e. on the stress field in the "Hertzian skin".

For rolling-element bearings it took some time before tribologic refinements on the contact mechanics of Hertz were introduced, i.e. after 1967 when P.H. Dawson [Ref. 26] had developed the aforementioned criterion of the "specific film thickness", or say of the Λ-ratio (see definition (1)). In essence this criterion accounts for the penetrating effects of the "roughness skin" on the EHD lubricant film, thereby causing the highly localized stresses on the tips of contacting asperities to be superimposed upon the Hertzian overall stress field.

For gears it took longer, i.e. till around 1914, before Hertz's maximum contact pressure was introduced as a criterion for pitting. But even prior to adopting Dawson's Λ-ratio gear designers felt a need for accounting for refining the Hertzian theory through other tribological criteria. These are typical of meshing tooth faces in that the sliding thereon occurring introduces frictional stresses as well as the concurrent frictional heating in the conjunctional zone of the tooth faces concerned. First of all, as early as 1942, the stress field due to the frictional stresses all by themselves was evaluated by F. Karas in Germany so that it could be superimposed upon the Hertzian stress field. Thereupon, in the late fifties, the flash temperature theory was applied in assessing the field of thermal stresses set-up by the non-uniformity of the

perature field in the "thermal skin", which usually is much thinner than the "Hertzian skin" albeit not as thin as the "roughness skin". In this connection it may here be observed that the thickness of each of these various skins may be put equal to a depth characteristic of the phenomenon concerned.

The late B.W. Kelly and R.P. van Zandt were presumably the first who, at the author's suggestion, numerically evaluated several such thermal stress fields, i.e. on the basis of the so-called thermal stress potential. Their results, unfortunately never published in full, corroborated the contention that in design calculation for pitting in fast-running heavy-duty gears the effect of the thermal stresses should not be ignored. For substantial treatments of these stresses the reader is referred to the 1967 paper of V.C. Mow and H.S. Chang [Ref. 27] and the recent one of B.J. Roylance et al.[Ref. 28].

Further, with gears running at the highest speeds nowadays attainable it may be derived from R.A. Burton's 1980 paper [Ref. 18] that still another thermally affected tribological effect may well have to be accounted for, i.e. that of the "thermal bulging" in the conjunctional zone of meshing tooth faces. Like the aforementioned thermal stresses the "thermal bulging" is caused by the non-uniformity of the flash temperature field in the "thermal skin". In the first place, the higher the speed and the consequent frictional heating of the gears concerned, the stronger the effect of thermal bulging on the profile, and thereby on the performance potentialities of both fully and partially separating thermo-EHD lubricant films. Last but not least, through some thermal feed-back process the thermal bulging might result, albeit only at rather high sliding velocities, in collapse of the EHD lubricant film through "implosion", thus causing scuffing (see the foregoing item no. 2.4.3 and Refs. 19, 20 and 21).

So, in the future development and design of gear transmissions towards higher power densities and as far as the pitting barrier is concerned, for the gears even more tribological refinements will have to be introduced than for the rolling-element bearings.

Let us now turn to certain fundamentals of the scuffing barriers. In passing it may be noted that in rolling-element bearings "smearing", which is related to scuffing, does not in general constitute as formidable a barrier as in gears.

Since the middle thirties it has been recognized that the contact temperatures, as generated by the frictional heating on meshing tooth faces, are even more influential as to still another tribologically critical phenomenon, the adhesive kind of wear called "scuffing". Thus it proved important to establish an evaluational method for these temperatures. The basic idea underlying the most generally adopted such method is that the contact temperature may be conceived to consist of two components to be superimposed upon each other. One component is the bulk temperature of the rubbing body to be considered whilst the other is the so-called "flash temperature", the latter being confined to the aforementioned "thermal skin" and attaining its maximum on the rubbing surface itself. In sufficiently severe duty where direct and intimate contact prevails the two rubbing surfaces may be expected to have one single distribution of contact temperatures in common. This can be justified on the conventional physical ground that there cannot be an interfacial temperature jump in the contact areas concerned, at least not in really intimate thermal contact. But in the other limiting cases where the two rubbing surfaces are fully separated elastohydrodynamically, the "contact" temperatures are interfacial ones of the lubricant film and either rubbing surface. Then there are two distributions of "contact" temperatures which will still show somewhat similar, but not identical trends along the lubricant film (see sub-section 2.3 for the question of a realistic thermal boundary condition for the thermo-EHD theory).

Already prior to World War II the author carried out numerous scuffing experiments in quite a variety of automobile transmission and gear oils, both straight mineral and E.P. ones. In interpreting these experiments in terms of contact temperatures as incipient scuffing he used two theories in conjunction, one being the flash temperature theory at is was first published already in 1937 [Refs. 29, 30 and 31], the other being the thermal-network theory published only in 1969 [Ref. 32]. Using electric immersion heaters for calibrating purposes, the latter theory proved fruitful in evaluating the individual coefficients of tooth friction at incipient scuffing. Therefrom the corresponding flash temperatures and the equally individual critical contact temperatures could be assessed for the various gear oils and critical operating conditions.

The major finding was that of the postulate about the constancy of the critical contact temperature at incipient scuffing, i.e. as a characteristic of each of the various non-additive mineral oils tested in conjunction with the gear steels employed [Refs. 33 and 34]. The underlying extensive experimental evidence on actual gears was destroyed owing to the war conditions in the author's country and thus could never be published. This is unfortunate since the scuffing experiments of others on disc machines have far from generally corroborated the present postulate. On the other hand, as has recently been pointed out in a private communication of the author's countryman, Dr. J.W. Polder, postwar scuffing experiments of others, again on gears, can be interpreted such that they do support the author's postulate. These are the experiments, possibly even more numerous then the author's prewar ones, although not covering as many kinds of gears as the author's, that were performed at Munich University of Technology, West Germany, under the supervision of the late Professor G. Niemann and his successor Professor H. Winter.

Last but not least, it would appear that satisfaction still persists among at least American designers of aerospace gear transmission after their long experience with designing gears along the lines of the still current AGMA Information Sheet No. 217.01, issued as early as 1965 by the American Gear Manufacturers Association [Ref. 35], and which is soon to reappear in an updated version. Since this design guide is based on the flash temperature theory in conjunction with the present postulate, it would appear that the latter may, at least with straight mineral oils, be upheld for practical purposes of gear design. However, the causes underlying the non-constancy of the scuffing temperatures of the various kinds of extreme pressure oils, have yet to be assessed more fully. Only thereafter will it prove possible to cast data about the scuffing performance of such chemically reactive oils into a form suiting gear design calculations.

In this connection it should be observed carefully that, of course, the constancy of scuffing temperature with straight mineral oils cannot be taken in any absolute sense, neither in the author's prewar gear tests nor in the later ones of others. But certainly through the former tests the author felt satisfied by the fact that the scatter then observed in these scuffing temperatures was much less then the one usual in

gear pitting tests, at least in terms of the uncertainties prevalent in the design stage as to predicting the power transmissible by the gears concerned.

It follows that, like in the conventional design calculations for pitting, it should come natural to gear designers to adopt a stochastic approach also in those for scuffing. For a fairly simple such approach the reader may be referred to the aforementioned AGMA Information Sheet [Ref. 35] and for a more elaborate one to that developed in the 1978 paper of the late P.M. Ku et al [Ref. 36] on the interpretation of their disc-scuffing tests in terms of gear-scuffing performance. In this connection it is worth mentioning that, on the basis of numerous exploratory investigations, the latter authors took great care in making the scuffing condition in their disc test as simulative as possible of those in gears.

Unfortunately, there is still a paucity of correlations for coefficients of tooth friction that have been generalized for the effects of influential quantities such as tooth load, certain geometric and kinematic characteristics of the toothing system concerned, certain material characteristics of the lubricant and the rubbing materials, not to forget roughness characteristics such as the plasticity index (for the material characteristics concerned, see sub-section 2.3 on thermo-EHD lubrication). Until sufficiently strict such correlations should have developed for a wide enough variety of kinds of gears and operating conditions, gear design calculations for scuffing will remain afflicted with uncertainties in the prediction of both the flash temperature component and the bulk temperature component of the contact temperature sought.

Despite the overwhelming amount of data on coefficients of friction to be found in literature about disc machine experimentation, correlations of the above-defined generalized kind are not even available yet for discs. In fact, the only kinds of correlation published so far are empirical ones in which at least a few of the aforementioned most influential quantities are only implicit in that they are concealed in fitted factors. The backwardness of this situation may from the present point of view be judged from data and discussions that may be found in papers such as those of D. Dowson [Ref. 37] and of P.M. Ku et al [Ref. 36].

The situation is even worse as regards sufficiently general kinds of correlation of coefficients of friction on tooth faces in mesh. So far next to nothing is known abôut the variation of the instantaneous coefficients of tooth friction from one meshing position to another. The only data available from tests on actual gears are those relating to some kind of average coefficient of friction that is assumed to be representative of the entire meshing cycle, as it may be derived through isolating the frictional power losses as accurately as possible from all the other such losses in the gear transmission concerned. Such an accurately enough isolation is a formidable task all by itself. Even though in his aforementioned prewar gear-scuffing tests the isolation through the thermal-network method proved successful, in his evaluations of critical contact temperatures at incipient scuffing the author had to make do with such averaged coefficients of tooth friction.

At least for aiming at sufficiently generalized correlations of tooth friction, if only of the ones averaged in the foregoing sense, a good start might presumably be made by following the lines indicated in figures 6 and 7 of the author's 1960 paper [Ref. 38]. Furthermore, considerable attention should yet be devoted to providing basic data about the various modes of heat transfer to be found in gear transmissions. Some of these modes are highly typical of such transmissions whilst apparently only one such mode, i.e. heat transfer by centrifugal fling-off of the lubricant from the rotating teeth has sofar been investigated fairly thoroughly [Refs. 14, 15 and 39]. All in all, the present basic data are indispensable for making the thermal-network theory really profitable, i.e. when it comes to predicting the bulk temperatures of the various elements in the gear transmission to be designed. After all, in the absence of these data one cannot very well predict the coefficients of tooth friction, these being affected by the bulk temperature of the meshing gears to be considered and which in turn determines the highly influential viscosity of the lubricant in the inlet cross-section of the EHD lubricant film.

4. CONCLUSIONS AND RECOMMENDATIONS

1. According as the trend towards higher power densities of gear transmissions continues their power envelopes tend to be increasingly afflicted tribologically with critical phenomena such as pitting and scuffing, which in turn are affected by various kinds of breakdown of the lubricant film.

2. A great many achievements in gear and bearing tribology have already proved valuable in contributing much to the aforementioned development towards higher power densities. However, as has been shown in the present paper, much remains to be done as to filling in quite a variety of missing links.

3. Furthermore, the trend concerned calls for an improved form of dissemination that sufficiently suits the needs of those responsible for the development and design of gear transmission. Otherwise too many potentially valuable efforts of tribologists may well prove insufficiently rewarding in that their ultimate application in actual practice experiences an undesirable delay. Then such efforts even run the risk of falling into oblivion so that, speaking with William Shakespeare, "Love's labour's lost". In any case, it is in the interest of all parties concerned to avoid such a situation.

4. Now, tribology has developed into so highly a multidisciplinary field that overspecialization of the great majority of tribologists provides a real obstacle to the aforementioned form of dissemination of knowledge and insight. This situation explains the urgency of the need for integrating tribo<u>logy</u> into gear design, i.e. by first developing it into tribo<u>technology</u>.

5. Hitherto the development of rolling-element bearings has well kept pace with that of gears. However, these two developments should continue to go hand in hand. Indeed, as gears and their lubrication are improved so that they can be made smaller for a given power throughput at some given input speed, the specific load capacity of the bearing should be increased accordingly since, thinking of the decreasing centre distances, less space can be made available for their assembly.

6. Once the aforementioned missing links should have been filled in, tribology, and particularly its integration into design, may be expected to provide important breakthroughs in the development of the technology of both gears and bearings to the high-tech level required for creatively increasing the power densities permissible in gear transmissions.

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14. Abstract

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- Experience of Tribological Aspects of Gear and Bearing Systems
- Future Design Trends and Problems
- Gear and Bearing Materials
- Lubricants

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